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7. Bona fide manufacturers, research students and persons interested in indigenous industries may consult books and periodicals in the Library at the notified hours.

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Compressed Air Plant

THE PRODUCTION, TRANSMISSION AND USE OF COMPRESSED AIR, WITH SPECIAL REFERENCE TO MINE SERVICE

BY

ROBERT PEELE

Mining Engineer and Professor of Mining in the School of Mines, Columbia University

FOURTH EDITION, REVISED AND ENLARGED

TOTAL ISSUE SIX THOUSAND

NEW YORK

JOHN WILEY & SONS, INC.

LONDON: CHAPMAN & HALL, LIMITED

1920

PREFACE TO THE FOURTH EDITION

In this edition a new chapter has been added, Chapter XXVII on "Measurement of Air Consumption." It contains data on the flow of air through orifices and short tubes, and a discussion of different appliances for measuring air at both low and high pressure including simple tank apparatus that can be readily improvised.

A note has been appended to Chapter XXV relative to some important air lift work recently done at a large mine in Mexico. The results of this work, which is still in progress as this edition goes to press, throw much light on the problems of raising water by air lift under heavy heads, and show the adaptability of the method to unwatering deep mines.

The author has availed himself of the opportunity to correct such typographical errors as have been found in the previous issue of the book.

R. P.

June, 1920.

PREFACE TO THE THIRD EDITION

SINCE the third printing (2d Edition) of this book, in 1913, there have been changes of some importance in the details of design of both compressors and rock-drills. Many of the illustrations have therefore been replaced in this edition by cuts from current drawings furnished by the makers, and the necessary changes in the text have been made.

In compressor design, the tendency during the last few years has been towards the larger use of "thin plate" air valves, and an increase in the size and consequent efficiency of the intercoolers of stage compressors. For all except very small machines, stage compression is now the rule. Illustrations and descriptions of certain of the older compressors and drills have been retained in this edition, even though their makers consider them obsolete. This is thought advisable, because, while the builders may be giving especial attention to new or modified designs, large numbers of the older machines are still in use, and will continue to be employed for a long time to come.

Among rock-drills, the old piston type has been little changed. Its field of work has become somewhat narrower, chiefly because of the improvements and larger use of hammer-drills for nearly all kinds of rock excavation.

For this edition a number of the chapters have been entirely rewritten and several have been expanded. As it was not desired to increase the size of the book, added space has been made by condensation and omission of old matter. The matter contained in the appendix to the 1913 edition has been incorporated in Chapter XX.

R. P.

NEW YORK, July, 1918

PREFACE TO THE SECOND EDITION

This edition has been revised and substantially enlarged. Among the principal additions are some 90 pages of text and 63 illustrations, relating to the construction and operation of rock-drills, coal-cutting machines and channeling machines. This material is contained in Chapters XX, XXI, XXII, and XXIII. The detailed records of work of machine drills, in Chapters XX and XXI, I believe, will be found useful. Most of the data has not before been in print.

In Chapter III the theory of the compression of air is presented in greater detail, together with its applications to the operation and performance of compressors. The deductions of the more important formulæ are also given, such as those used for calculating the horse-power required for single- and multiple-stage compression. In this connection I desire to acknowledge the kind assistance of Professor Charles E. Lucke, of Columbia University, and Professor H. J. Thorkelson of the University of Wisconsin. To Dr. Lucke my thanks are due for the use of his valuable, and hitherto unpublished, notes relating to the work cycles of air compression, with and without clearance. I would call attention also to the records of compressor tests in the latter part of Chapter X. These comprise a few typical tests, selected from a large number recently made by Mr. R. L. Webb, Mechanical Engineer, on compressors of different kinds in a well-known Canadian mining district.

Other new material has also been added, relative to the piston clearance of the air cylinders of compressors, and the ratio of inlet valve area to cylinder area. Numerous minor additions to the text have been made, together with corrections and alter-

ations where required. The new matter aggregates some 135 pages of text and 87 illustrations. Many of the illustrations have been furnished by the respective makers of the machinery, to whom credit is duly given. In preparing this revision I have kept in mind certain kindly criticisms and suggestions received from readers of the first edition. R. P.

NEW YORK, June, 1910.

PREFACE TO THE FIRST EDITION

THE increasing use of compressed air makes the subject of interest to practitioners in nearly all branches of engineering. Besides its more important power applications, such as the operation of rock-drills, air brakes, riveting machines, and railroad switching and signalling systems, the uses of compressed air are numerous in many minor branches of mechanical engineering, in caisson work and the construction of subaqueous foundations, and in manufacturing industries, chemical works, etc., where it serves a multitude of purposes entirely distinct from that of the transmission of power.

A realization of the breadth of the field has suggested that a book may be acceptable, addressed especially to those who are engaged in mining, tunnelling, quarrying, and other work involving the excavation of rock, with its concomitant operations. While the literature bearing upon this branch of compressed-air service is by no means small, it is for the most part scattered through the technical periodicals and transactions of engineering societies, and therefore not readily accessible to those who are out of convenient reach of engineering libraries. I am aware that little that is new can be said on this subject, and in writing the book I have availed myself freely of existing sources of information.

In the first part, I have endeavored to present a view of current practice as to the construction and operation of compressors.

Portions of the subject are dealt with at some length, for example: the types of compressor suitable for different kinds of service, heat losses occurring in air compression, and the various forms of valves, valve-motions, and governing and unloading

mechanisms, that constitute prominent features of modern compressor practice. A brief review is given of a few of the fundamental principles of air compression, but my intention has been to present only enough of the theory to make intelligible the formulæ employed for the ordinary calculations of the power and capacity of compressed air plant, together with the questions concerning temperature changes, as affecting the production and use of compressed air. Many details of the design of compressors and proportions of their parts have been omitted, since these fall properly within the province of the mechanical engineer.

The second part is devoted to the applications (largely to mine service) of compressed-air transmission of power, including machine drills, pumps operated by compressed air, and mine haulage by compressed-air locomotives.

Many of the illustrations are reduced or adapted from working drawings kindly supplied by compressor builders. Others have been taken from catalogues of compressed-air machinery and from technical periodicals and books dealing with the different types. The origin of these has been stated in nearly every instance. My thanks are due to several of the technical journals, especially *Compressed Air Magazine* and *Mines and Minerals*, for many suggestions and in some cases for passages extracted either in substance or verbatim, from articles therein contained. For any important use or adaptation of published material, permission has been asked and obtained, and frequent references are given in foot-notes or in the body of the text. I also wish to acknowledge my indebtedness to Mr. Frank Richards' book on "Compressed Air," from which I have derived substantial assistance.

ROBERT PEELE.

SCHOOL OF MINES, COLUMBIA UNIVERSITY,
NEW YORK, May, 1908.

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PRODUCTION OF COMPRESSED AIR

CHAPTER I

INTRODUCTION

One of the most important applications of the transmission of power by compressed air is the driving of machine rock-drills; and to the necessity of providing for these drills a power medium suitable for use in mines and tunnels has been due, more than to any other cause, the development of the modern air compressor.

The time which has elapsed since the beginnings of this branch of engineering is short. The first percussion rock-drill operating independently of gravity, was invented in 1849 by J. J. Couch, of Philadelphia. Though used only experimentally, it embodied the principal mechanical features of the modern machine-drills, which have had such a striking influence in mining and tunnelling. Couch's machine, together with its immediate successors, such as the Fowle drill (1849–51) and the Cavé (Paris, 1851), were steam-driven and therefore unsuitable for underground work. In 1852, the physicist Colladon proposed the use of compressed air for operating rock-drills, in connection with the driving of the Mont Cenis tunnel, in the western Alps. His idea was developed by Sommeiller and others between 1852 and 1860, and in 1861–62 an air-compressor plant was first used successfully

at that tunnel. It was driven by water power and furnished air for ventilation as well as for the drills. The early air compressors were crude in design. Sommeiller's first plant, though of large size, had some resemblance in principle to the old hydraulic ram, possessing no moving parts except the valves. Steam-driven piston compressors, as the Dubois-François, and more or less similar to some of the wet compressors still in use in Europe, soon made their appearance. The first compressors built in the United States were the Burleigh, employed at the Hoosac Tunnel, on the Boston and Albany Railroad, in 1865–66. The Norwalk, Clayton, and Rand compressors were among the earlier makes in this country.

In Europe, the Mont Cenis tunnel, about eight miles long and completed in 1871, the first connecting link through the Alps between the railway systems of France and Italy, was the field where were solved on a large scale many problems of compressed-air production and use. Sommeiller there laid the foundations of new practice, by which that great work was successfully completed. From 1857 to 1861 the tunnel headings had been progressing slowly and in the face of great difficulties. Drilling was done by hand labor and blasting by black powder, the average advance for this period, in each of the two headings, being only about 1.5 ft. per day. At this rate, granting that the work could have been finished at all by the means employed, over 40 years would have been required to connect the headings and years more to complete the enlargement to full section. With machine drills, the average speed of advance in each heading rose to 4.75 ft. per 24 hrs. and later, when dynamite was introduced, to a little over 6 ft.; this average being maintained for a period of 6 years.

Machine drills did not make their way into mining to any extent for some years after their successful application to tunnel-driving. It is difficult now to name the mining district in this country where they were first used, but their most important trial was in the Calumet and Hecla copper mine, Michigan. After strong opposition from the miners, the Rand drill was introduced there in 1878, and the value of machine drilling was

soon demonstrated by decreased costs of drifting and stoping and higher speeds of advance.

Compressed air has now a wide application in various branches of mechanical engineering. In this book it is intended to deal only with its production and uses for mining and tunnelling operations. Its two rivals in these fields of work are steam and electricity.

As compared with steam, compressed-air transmission of power is useful and convenient for three reasons: *first*, its loss in transmission through pipes is relatively small; *second*, the troublesome question of the disposal of exhaust steam underground is avoided; *third*, the exhausted air is of some assistance in ventilating the working places of the mine. In large mines, where steam may be carried thousands of feet, down shafts and through lateral workings, for operating pumping engines, etc., the disadvantages attending its use are apparent; condensation is serious, even when the piping has good non-conducting covering, and the efficiency becomes abnormally small. Furthermore, aside from the heat produced by the use of steam, it is rarely feasible to employ efficient condensers for underground engines other than pumps, because of the difficulty of obtaining condensing water. If the exhaust be discharged into the mine workings, even though these are large and well ventilated and the volume of the exhaust steam comparatively small, the temperature and quantity of moisture in the air is considerably increased. Deterioration of the timbering is hastened, the roof and walls of the workings are often softened and slacked off, and the mine atmosphere is rendered oppressive and unwholesome. The presence of hot steam pipes in confined workings, or in the narrow compartments of shafts, is also objectionable.

Even with use of the best non-conducting covering the condensation loss in long steam lines greatly reduces the effective pressure at a distant underground engine and very uneconomical working is the result. In conveying steam several thousand feet the pressure may be reduced to half the boiler pressure, or less. Thus, in the case of a pump situated 2,000 ft. from the boiler and using 200 cu.ft. of steam per minute at a boiler

pressure of 75 lbs., with a 4-in. mineral-wool-covered pipe, the effective pressure at the pump would be only about 58 lbs.; or, with a poor covering, like some of the asbestos lagging often used, it might be as low as 35 lbs. In compressed-air transmission, on the other hand, the reduction of pressure for the same volume of air, size of pipe, and initial pressure, would be 9.3 lbs., giving a terminal pressure of 65.7 lbs. However, as the speed of flow in pipes for economical transmission is greater for steam than for air, a comparison based solely on piping of the same diameter cannot justly be made. In the above example, if the diameter of the pipe were smaller the gain in reduced radiation would outweigh the increased frictional loss, and the net loss would be diminished. Since the frictional loss varies inversely, and the loss from radiation directly, with the diameter, the size of the steam pipe can be so proportioned as to produce a minimum loss under given conditions. With compressed air the case is different, since the question of radiation is eliminated. If the pipe diameter be increased to 5 ins. the loss of pressure, or head required to overcome friction, is reduced to 2.8 lbs., and increasing the distance to one mile it would be only 7.4 lbs. Furthermore, as against the increased cost of the larger air pipe, there is the expense of the non-conducting covering necessary for steam transmission.

Thus, compressed air may be conveyed long distances with but small loss of pressure; it is always ready to do its work, and, aside from leakage of pipes, which is preventable, it suffers no loss of power when not in actual use. For performing work *intermittently*, at a distance from its source, it is therefore particularly valuable, because the air pressure is maintained nearly constant during intervals of work, without further expenditure of power. With steam transmission, power is continually dissipated by radiation, and a steam engine, when stopped for any length of time, loses much of its normal working temperature and becomes a receptacle for water of condensation.

Though compressed air is employed in mining mainly for operating machine drills, it is used also for underground hoists and pumps, and sometimes for mechanical coal cutters, in both

bituminous and anthracite mines. Compressed-air locomotives in mines and tunnels exemplify its capacity for storing power, in contradistinction to its function as a power transmitter. The introduction of machine drills has facilitated the driving of railroad and mine tunnels, which otherwise would have been greatly delayed or completed only with difficulty. Had compressed-air power, together with the high explosives, not been available, it is doubtful whether the great tunnels in the Alps and elsewhere, and the numerous long mine tunnels driven in recent years in this country, would have been at all practicable.

Without reviewing in detail the comparative merits of electricity and compressed air, it may be pointed out that the application of electricity for transmitting power in mines has increased enormously in importance since about 1888. The peculiar requirements of mine service have been in most cases successfully met by modifications of standard forms of electric apparatus. Both means of power transmission possess characteristics which adapt them particularly for underground work. But, although electricity rivals compressed air in nearly all branches of mine work, their spheres of usefulness are not identical and the field is broad enough for both. Though it is sometimes stated that the first cost of an electric plant is lower than that of an equivalent compressed-air plant, there is actually but little difference between the costs of the plants themselves. For short-distance transmission of a given power an electric conductor line costs much less than compressed-air piping; but the cost of the electric line increases as the square of the distance, while that of the pipe line increases only as the first power of the distance. It is in the greater efficiency of generation that electric transmission has the most marked advantage.

In one direction only has electricity failed hitherto to meet every requirement. While compressed-air drills, though far from being economical machines, nevertheless admirably fulfil their purpose, no satisfactory electric rock-drill has yet been produced. It is to be hoped that a solution of this problem may be found. The Temple-Ingersoll "electric-air" drill, brought out about 1902, is an ingenious machine, but not an

electric drill in the ordinary meaning of the term. It is a combination of a compressed-air drill of special design, operated by a small, electric-driven compressor. As there is no exhaust, an incidental advantage of the ordinary air drill is missing, namely, that of assisting somewhat in ventilating those places where ventilation is most needed. This, together with such minor uses of compressed air as the cleaning of drill holes preparatory to charging, and driving out the smoke of blasting from working places, renders it doubtful whether, for underground mining, electric drills of any kind can supersede entirely those operated by compressed air. Given the necessity for a compressed-air plant for rock-drills, as is the case in most metal mines, it may often be more advantageous to provide the additional compressor capacity required for driving underground pumps, hoists, and other machines as well, than to install a separate plant for generating electricity.

It has long been customary to consider compressed air as a mode of power transmission respecting which the questions of convenience and expediency are more weighty than the attainment of a high degree of efficiency. But, as the principles of air compression have become better understood, a substantial improvement has taken place in compressor design, the installation of pipe lines, and the operation of machines using compressed air. The consequences of overloading a compressor, and thereby driving it beyond its proper speed, are comprehended by every intelligent master mechanic as being wholly different from those due to overloading a steam engine. The results of leaks in air pipes, and of using air mains of too small a diameter, are also understood. By better practice in the production, transmission, and use of compressed air a higher total efficiency is now realized than formerly was thought possible.

CHAPTER II

STRUCTURE AND OPERATION OF COMPRESSORS

An air compressor consists of a cylinder in which atmospheric air is compressed by a piston, the driving power being derived from a steam engine, water-wheel, or electric motor. The air-cylinder is usually double-acting, with inlet and delivery valves in each head. The air is compressed by the advancing piston, while, in the simplest compressors, the decrease in pressure, or, as it is commonly termed, the tendency to a vacuum, behind the piston causes the inlet valves to open under atmospheric pressure, thus allowing air to flow into the cylinder. At each stroke a certain volume of compressed air is forced out through the discharge valves, into a pipe leading to a reservoir or receiver, whence the air enters the transmission pipe or main.

No single classification of air compressors can be made sufficiently comprehensive to present all of their salient features. Three bases of comparison suggest themselves. *First*, the structural characteristics of compressors regarded purely as engines; *second*, the mode of dealing with the heat produced during compression; *third*, the numerous types of air valves and valve-motions. The first classification is given here, the others being taken up respectively in Chap. IV–VI and VII–IX. Air-brake and gas compressors, vacuum pumps and other special air-compressing machinery are omitted, as this book deals only with compressors which are applicable to mine or similar service. (A classification including compressors for nearly all kinds of service, as built by an important American maker, is given at the end of this chapter.)

First Classification, taking the steam-driven compressor as the type form:

1. **"Straight-line" Compressors.** The steam and air cylinders are set tandem on a common piston-rod. There is a pair of fly-wheels, one on each end of the crank-shaft, driven by a single connecting rod, or, in some designs, by outside connecting-rods from a cross-head between the cylinders. (Figs. 1 to 4.)

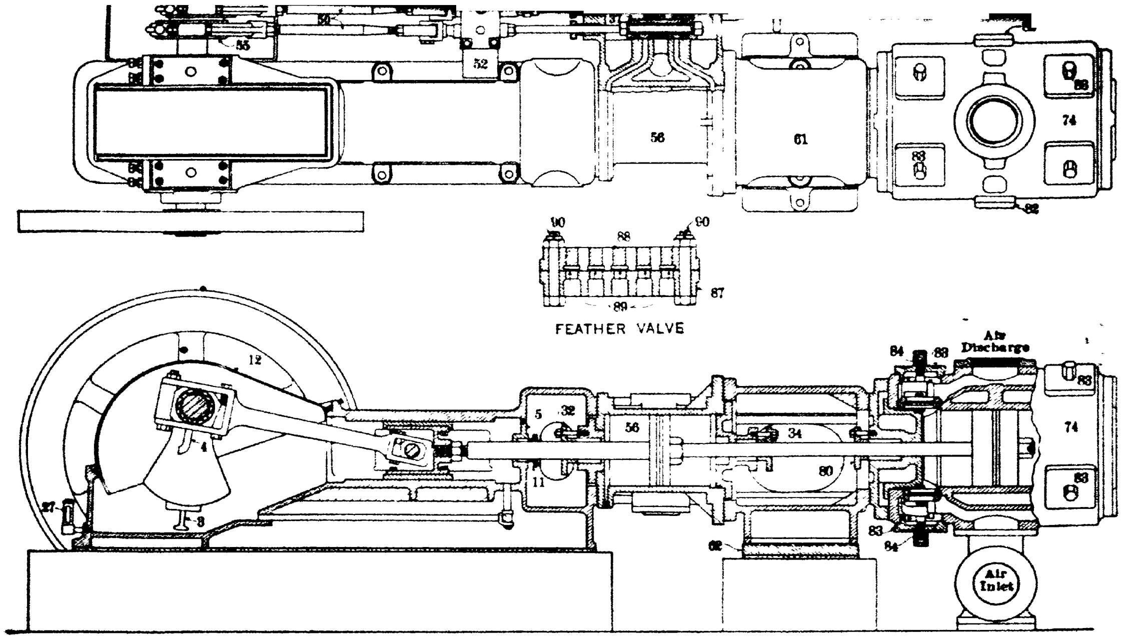
2. **Duplex Compressors.** Two engines are placed side by side, each consisting of tandem steam and air cylinders, with their cranks set at 90° on a common fly-wheel shaft. Each side of the duplex is in effect a straight-line compressor. The steam cylinders may be simple or compound; the air cylinders single or staged. (Figs. 9–14.)

3. **Compressors with Compound Steam Ends.** (*a*) Duplex, horizontal, cross-compound; a single-stage air cylinder being set tandem to each steam cylinder. This form is now rare. The considerations leading to the compounding of the steam end make it desirable to use stage compression. Nearly all large, steam-driven compressors are now of the duplex, cross-compound, two-stage type (Figs. 9–14). (*b*) Vertical compound; the air cylinders being placed above the steam cylinders. This also is an unusual design. Some large King-Riedler compressors,* up to a capacity of 8,000 cu.ft. of free air per minute, have been built for South African mines. In Great Britain vertical compressors, both large and small, have had considerable vogue in recent years. Prominent among them are those of Belliss & Morcom, Peter Brotherhood Co., Alley & MacClennan, and Robey & Co.

The chief advantage of vertical compressors is the saving of floor space, which is rarely of consequence at mines. Disadvantages are the relative inaccessibility of the working parts, as in Figs. 26, 30, and hence the difficulty of proper adjustment, maintenance and repairs.

4. **Stage Compressors,** in which the air cylinders are compounded. The air end may be of the double-, triple-, or quadruple-stage type, according to the air pressure to be produced. Stage compressors are now built by nearly all makers,

* *American Machinist*, Oct. 16, 1902, p. 1475.



3. Oil splasher.
4. Oil scoop.
5. Oil head.
11. Oil-head stuffer.
12. Oil guard.
27. Oil gage.
32, 34. Front and rear steam cylinder-head stuffers.
36. Expansion valve-rod stuffer.
37. Main steam slide valve.
38. Expansion slide valve.
50. Eccentric rods for main and expansion valves.
55. Steam eccentric.
56. Steam cylinder.
64. Governor pulley.
69. Handwheel for adjusting expansion valve (38).
74. Air cylinder.
80. Front air-head stuffer.
82. Hand-hole cover.
83, 84. Valve cover and screw.
87. Valve seat.
88. Valve guard.
89. Valve strip (see Figs. 62, 63, Chap. VII).
90. Ream bolts and nuts.

FIG. 1.—Laidlaw-Dunn-Gordon Straight-Line Compressor, 12″×13″×14″ (Feather Valves).

and are the most important class.* (*a*) Straight-line form, as in (1). These have two-stage air ends, some with compound steam ends also. Fig. 6 shows a Norwalk, and Figs. 7, 8 a Sullivan compressor, both having compound steam and two-stage air cylinders. Fig. 5 is a two-stage compressor with simple steam end. (*b*) Duplex, simple steam end, with two-stage air cylinders, a type now rarely built. (*c*) Duplex, cross-compound steam end, with two- to four-stage air cylinders. Those of more than two stages are for special high-pressure service, as for compressed-

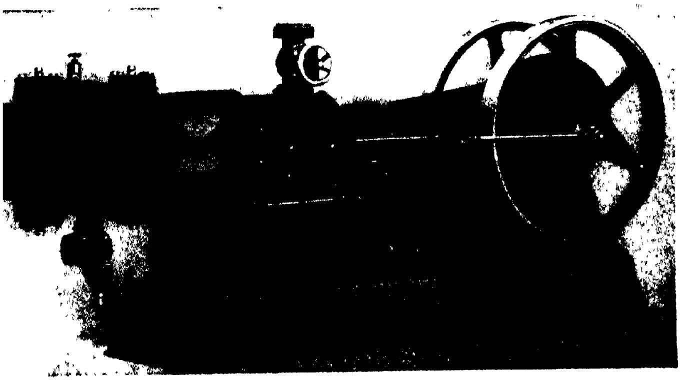


Fig. 2.—Ingersoll-Rand Straight-Line Compressor, Class "FR-1".

air locomotives. Figs. 9, 10, and 11 show the latest type of Ingersoll-Rand cross-compound, two-stage compressor, with piston steam valves; similar compressors of smaller capacity are fitted with Meyer slide valves. Figs. 12, 13 are of two types of Laidlaw-Dunn-Gordon compressors, the former having Corliss inlet and poppet discharge valves for the air end, the latter being a newer design, with "feather" air valves. Fig. 14 shows the latest type of Allis-Chalmers compressor. For illustrations of 3- and 4-stage compressors, see Chap. XXVI.

* The Norwalk Iron Works Co was the pioneer in the field of stage compression, having begun in 1880–81 to build this type of compressor for ordinary service.

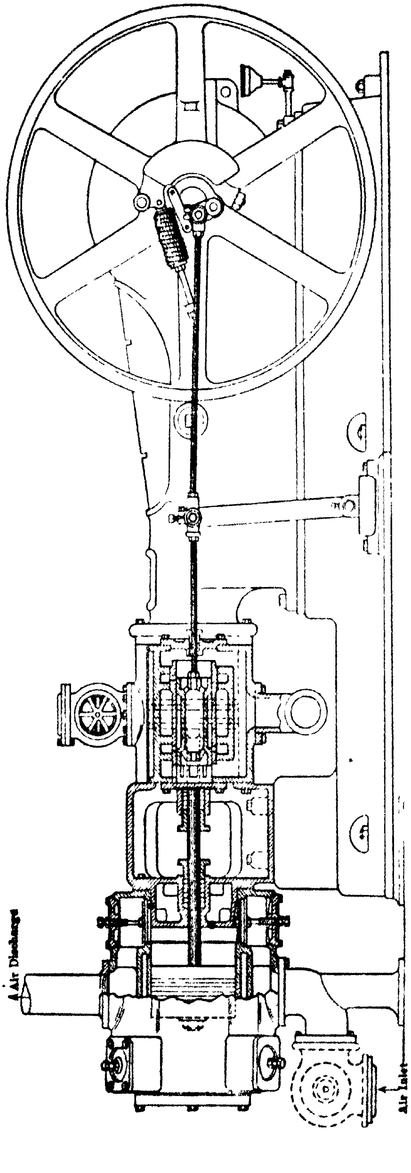


FIG. 3.—Ingersoll-Rand Straight-Line Compressor, Class "FR-1".

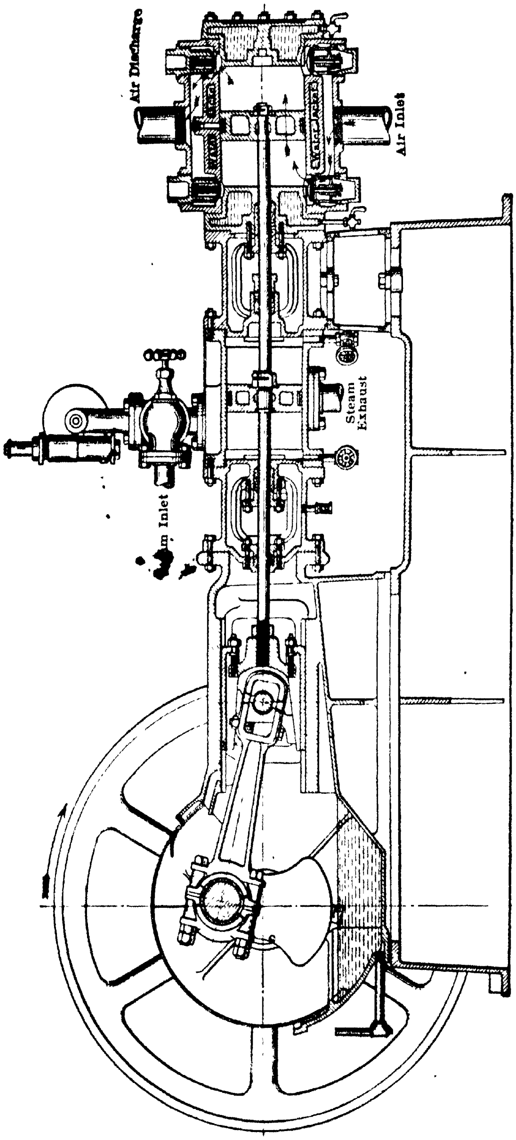


FIG. 4.—Sullivan Straight-Line Compressor, Class "WA-5."

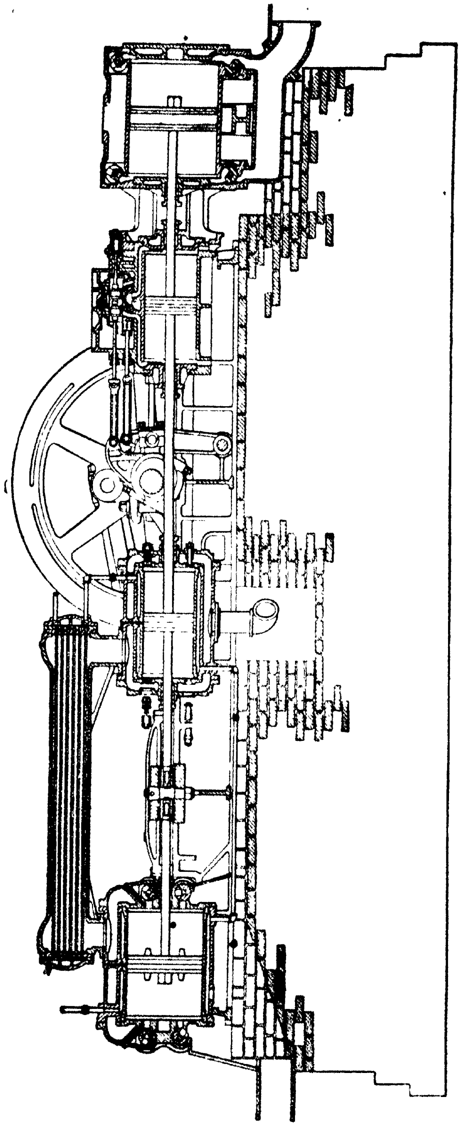


Fig. 6.—Norwalk Compound, Two-Stage Straight-Line Compressor.

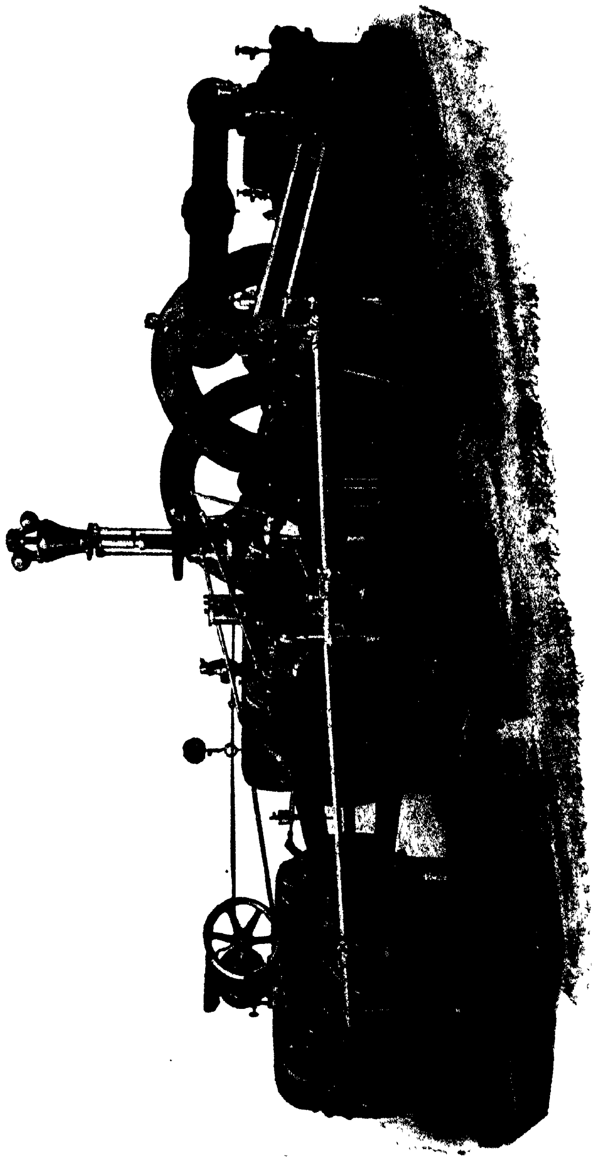


FIG. 7.—Sullivan Corliss Compound, Two-Stage, Straight-Line Compressor. Class 'WC,' 16" and 28"-14½" and 24"×24" Cylinders.

FIG. 9.—Ingersoll-Rand Compound, Two-Stage Compressor. "Imperial" Type, Class "XPV-3"

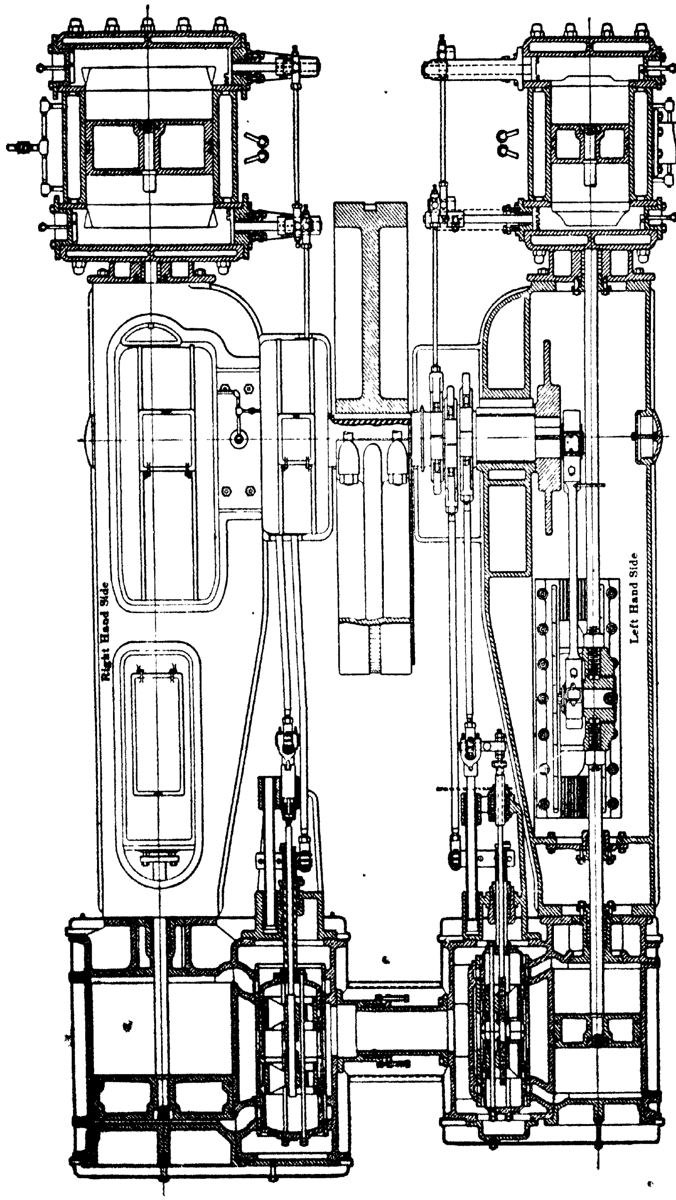
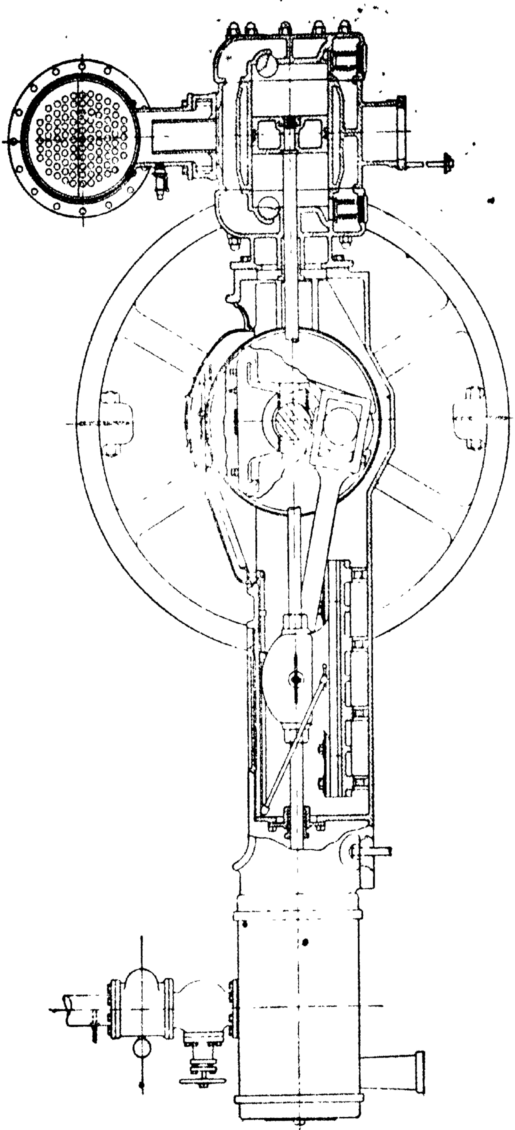


FIG. 10.—Ingersoll-Rand Compound, Two-Stage Compressor, "Imperial" Type, Class "XPV-3"



ELEVATION, HIGH PRESSURE SIDE

FIG. 11 —Ingersoll-Rand Compound, Two-Stage Compressor, "Imperial" Type, Class "XPV-3."

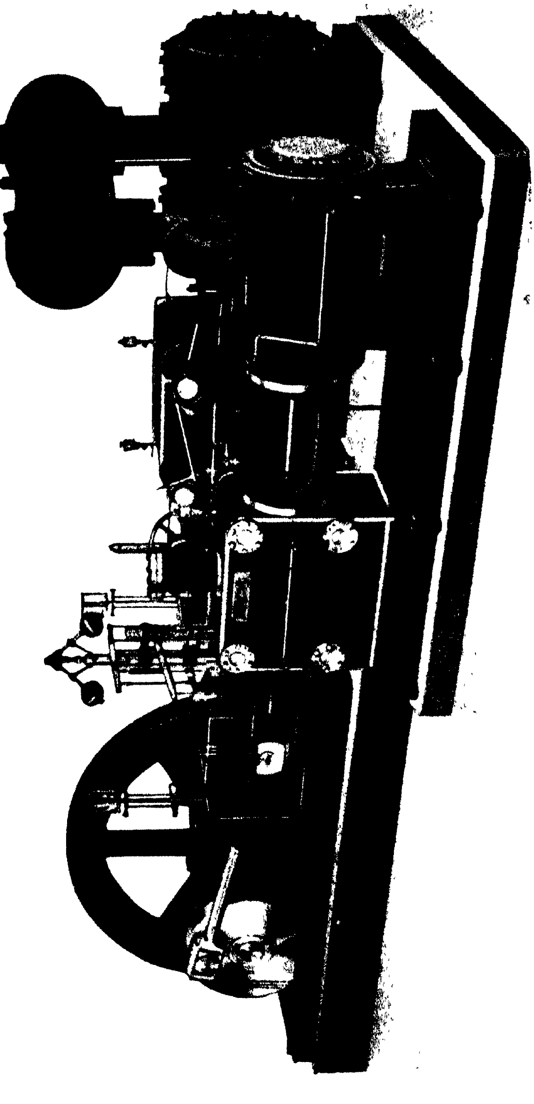


FIG. 13.—Laidlaw-Dunn-Gordon Duplex, Cross-Compound, Two-Stage Compressor, with "Feather" Air Valves.

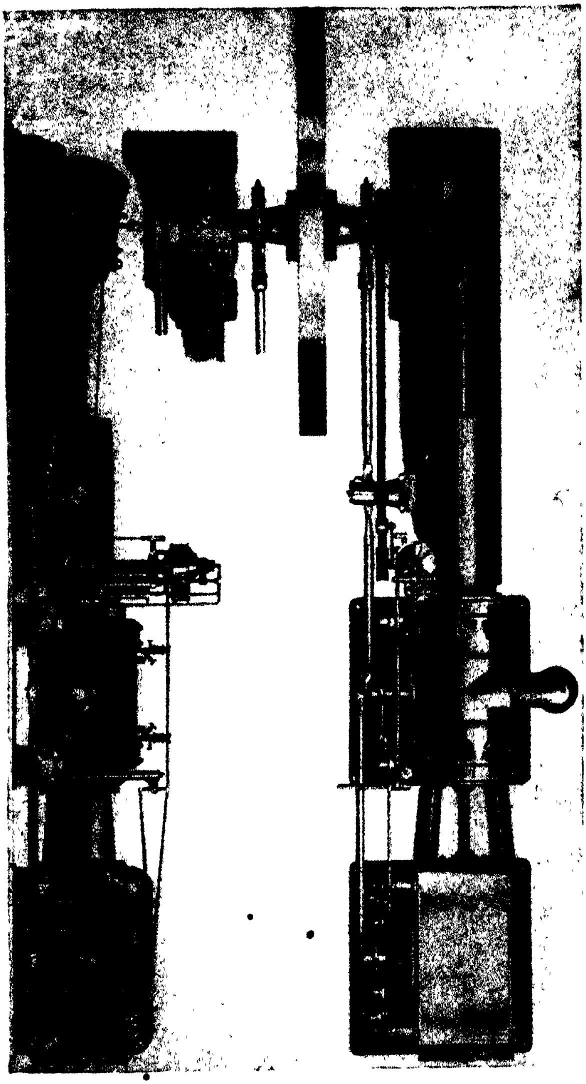


Fig. 14.—Allis-Chalmers Duplex, Cross-Compound. Two-Stage Compressor. (Corliss inlet and poppet discharge valves.)

Among well-known British compressors of this type are those of Walker Bros., Wigan, and Robey & Co., Lincoln. The Walker compressor is horizontal, duplex, two-stage, with simple or compound steam end, and is made of capacities from 1,100 to 9,400 cu.ft. free air per min. Robey & Co. make high-speed, vertical (see Fig. 30), as well as low-speed, horizontal compressors. The machines of both of these builders have large disk, or plate, air valves (Chap. VII). The Riedler compressor, formerly made by Fraser & Chalmers, Erith, England, has been largely used for mining work. For its unique air valves, see Chap. IX.

As based on structural characteristics, compressors may also be classified as: (*a*) *Direct-driven* by steam, electricity, or water-power—the motor end being directly connected with the air cylinders. Among water motors the bucket or impulse wheels are best adapted to this service; (*b*) *Belt-driven* from independent motors: steam-engines, water-wheels, or electric motors. These are in common use for mine and other service. Chain-driven and direct-connected compressors are also occasionally employed. (Figs. 28, 29.)

So-called "half-duplex" compressors consist of either the right- or left-hand half of a duplex compressor, an extended crank-shaft and out-board pillow-block being provided temporarily. If a comparatively small quantity of air is needed for a time—as during the development of a mine or the sinking of a shaft—a half-duplex may be installed at first, the second half being added later. The capacity is thus readily doubled at moderate cost.

Comparison of Types of Compressor

The **straight-line compressor** is largely employed for small or medium size plants, or for temporary service. It is compact, strong, and self-contained, being carried on a single bed-frame and requiring a relatively inexpensive foundation. The floor space occupied is much less than for the duplex form.

While useful for moderate air pressures and fairly constant

loads, the straight-line compressor is not capable of operating with the steam economy essential in large plants: nor is it self-regulating at much less than, say, 40% of its full load. These compressors are usually made of capacities from the smallest up to 1,700 or 1,800 cu.ft. of free air per minute, the last-named sizes developing 275-300 H.P.

The **duplex compressor** is preferable to the straight-line for large plants. It is better adapted to varying loads, arising from differences of air pressure, because the resistance is more uniformly distributed throughout the stroke. Having quartering cranks it will run at very low speed without stopping on a center; it is self-regulating and capable of dealing economically with a range of load down to less than one-quarter or one-third of its normal. As a rule, the friction loss (total H.P. consumed by engine friction) is no greater and is often less than that of a straight-line of the same capacity. For large compressors, in good order, this loss may be not over 5-7%.* While these figures are sometimes equalled by the best straight-line compressors, the loss in the latter is generally higher.

The Corliss type of engine is frequently used for driving large duplex compressors, as its valve gear is well adapted for dealing with variations of air pressure. Corliss valves have been largely employed in the past, for the air as well as the steam cylinders. Since about 1913 air valves of the thin-plate or "feather" type have been adopted by several well-known makers. (See Figs. 1, 2, 3 and 24; also Chaps. VII, VIII and IX.)

The foundation of the duplex compressor is necessarily more expensive than that of the straight-line, and to maintain perfect alinement must be substantially built. Each pair of cylinders are connected by trunk-frames or tie-bolts. A complete girder-frame (Figs. 1, 2, 9, 12, 14) prevents any possibility of movement. The tandem steam and air cylinders on each side

* See an article by J. Parke Channing, in *Mines and Minerals*, May 1905, p. 475, containing the results of an efficiency test on a 300-H.P., compound, two-stage Nordberg Corliss compressor, at the Burra-Burra mine of the Tennessee Copper Co. Its total efficiency was found to be 78.1%. The horse-power consumed by friction was only 5.2% (see p. 149).

are best placed far enough apart to prevent the same portion of the piston-rod from passing alternately into each stuffing-box, because: *first,* as the piston-rod is apt to wear differently in the two stuffing-boxes, it becomes difficult to keep them well packed and tight; *second,* the steam and air piston-rods are often in separate parts, coupled between the cylinders. This is convenient in dismantling the compressor for repairs; also, the air valves, when of the poppet form and in the cylinder head, are more accessible.

Compressors with Compound Steam Cylinders. The economic advantages of compounding the steam end are greater than in ordinary engines: *first,* because the conversion of power from one form to another is necessarily attended by loss, and should be conducted as efficiently as possible; *second,* because, as shown below, air compression involves unfavorable load conditions. A steam saving of, say, 20% may be readily attained by using a good condenser, thus getting the full expansive power out of the steam, and by avoiding loss of power due to imperfect speed regulation and consequent blowing off of air at the safety valve.

Stage compressors. It is now recognized that even for pressures of 75-90 lbs., as commonly employed for machine drills, a saving in steam consumption can be realized by stage compression. In mountain regions, where so much mining is carried on, its advantages are still greater than at sea-level (Chap. XIII). The duplex form, with both steam and air ends compounded, exemplifies the highest type. There is no material increase in the number of moving parts, except valves; the greatest range of steam expansion is obtainable, because the work done in the air cylinders is more nearly equalized, and the compressor may be made self-regulating over its entire range of load. Thermodynamically, the efficiency of stage compression depends largely on the proper use of water-jackets for the air cylinders, and the size and design of the intercoolers. (Chap. VI.)

The Operation of Steam-driven Compressors involves conditions which do not obtain in ordinary steam engines (see Fig.

15). At the beginning of the stroke the air in front of the piston is approximately at atmospheric pressure. As the piston advances the pressure at first increases slowly, and then very rapidly to its maximum. The power developed in the steam cylinder, when working as usual with a cutoff, is in the reverse order. The initial steam pressure may be even lower than the final air pressure, though the mean effective pressure in the steam cylinder is greater than the mean effective in the air cylinder. For example, with an initial steam pressure of 60 lbs., air may be compressed to 80 lbs. or more. This is due to the use of heavy fly-wheels and reciprocating parts, which store up the surplus power in the early part of the stroke, and give it out toward the end. The consequent lack of smoothness in the

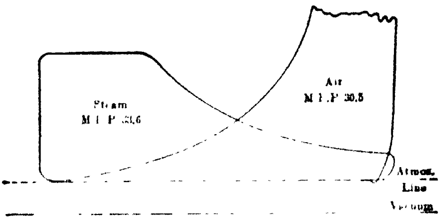


FIG. 15.—Combined Air and Steam Cards, Straight Line Compressor.

running of compressors is especially noticeable in the simple straight-line type, which, when the air in the receiver is at maximum, often comes almost to a standstill and barely turns over the centers. It would thus appear that only a small ratio of expansion in the steam cylinder could be employed; but the difficulty is met by the inertia of the fly-wheels, and the cylinders of even simple compressors can be proportioned for a steam cutoff at ⅜ to ½ stroke. In the duplex type, power and resistance are more nearly equalized, the most favorable distribution being attained in cross-compound, stage compressors.

Steam Valves are of a number of forms: plain slide-valve, with or without Meyer adjustable cutoff, Corliss, drop valve, and balanced piston valve.

In most simple straight-line compressors the steam cylinder

has an adjustable cutoff valve (Fig. 1). This valve (38) is composed of two parts on a right- and left-hand threaded stem, and, by moving on top of the slide valve, controls ports in the latter through which steam is admitted to the main ports. It is operated by a separate eccentric on the fly-wheel shaft, and by the hand-wheel (69), outside of the valve chest, may be regulated without stopping the compressor, according to the varying receiver pressure. By manipulating this valve the engineer can prevent the compressor from stopping on a dead center, notwithstanding variations in air pressure (see also Fig. 12).

The Corliss valve is used by a number of makers, chiefly for large engines, both straight-line and duplex, and is well adapted for compressor service (Figs. 6, 7, 8 and 14). The double-ported steam valves in Fig. 8 enable the compressor to run at a higher speed.

Piston steam valves have recently been adopted by the Ingersoll-Rand Co. for its straight-line and Imperial "XPV" compressors (Figs. 2, 3, 9, 10). They are balanced, of the telescopic type. Their accompanying cutoff valves, one for each end of the cylinder, are right- and left-hand threaded on the cutoff valve stem, which telescopes through the main valve stem. Steam enters the interior of the main valve, and passes out through the ports near the ends, being exhausted by the ends of the valve. This design requires only two, instead of four, ports, thus reducing clearance and condensation surface. The main valve is in two parts; the ends, which separate live steam from the exhaust, are cast integral with each half, thus avoiding the need for steam-tight joints at these points. As steam exhausts past the ends, the valve covers and packing are under exhaust pressure only, thus decreasing leakage, and subjecting to low temperatures the exterior walls of the chest. This valve is well suited to high steam pressures and the use of superheating.

Drop valves are adapted to large compressors working with high steam pressure and superheat. This valve is a double-ported poppet, lifted by a cam and closed by a spring. It is nearly balanced, and, as dash-pots are not required, the running

speed of the compressor may be higher than for Corliss engines. Both ports and steam passages are large. The point of admission to the high-pressure valves is fixed, but an increase of air pressure beyond the predetermined point causes an earlier cutoff; the reverse conditions, a later cutoff. The low-pressure admission valves and all exhaust valves are set to give constant points of admission and cutoff.

Proportions of Cylinders. A short stroke is conducive to economy in compression, as well as the attainment of a proper rotative speed. It is of especial importance in simple straight-line compressors, because the power and resistance are then more nearly equalized. With a long stroke the piston would travel some distance under an increasing resistance; then, after the discharge valves open, it would complete its stroke under a uniform resistance, while adding nothing to the amount of useful work. But the loss of volumetric capacity due to piston clearance is less for a long than a short cylinder of the same diameter. In single-stage, slide-valve compressors, the usual ratio of stroke to diameter of steam cylinder is $1\frac{1}{2}:1$ or $1\frac{1}{4}:1$. In some recent designs, the stroke and diameter are nearly equal, while in duplex Corliss compressors are found such variations in the proportions of steam cylinders as: 12×30 ins., 14×42 ins., 20×42 ins., and 30×60 ins.

The relative diameters of the air and steam cylinders depend obviously on the steam pressure carried and the air pressure to be produced. At mines there is usually but little variation in these conditions, except for operating compressed-air locomotives (Chap. XXVI). For rock-drills, the air pressure is generally from 70–90 lbs. The applications of compressed air for manufacturing purposes have so multiplied that some builders furnish compressors to produce pressures of from 10–120 lbs. per sq. in.

Compressors Driven by Gasolene Engines have recently been introduced. Fig. 16 shows a small portable outfit, with the compressor short-belted to the engine. It is useful for prospecting, quarrying, and general surface rock excavation. Figs. 17 and 18 show straight-line gasolene-driven compressors.

They can be operated also on fuel oil and low-grade distillates. The power end of the compressor in Fig. 17 is of the valveless, two-cycle, low-compression type, without electric firing, and having the water supply and fuel injection to the combustion chamber positively governed. This compressor is built in four sizes, with air cylinder capacities (piston displacement) of 70–290 cu.ft. per min. (10–46 H.P.). It is also made in portable form, on wheels, and semi-portable, on skids.

Compressors Driven by Water Power. Impulse wheels, as the Pelton, Knight, or Risdon, are best adapted to this

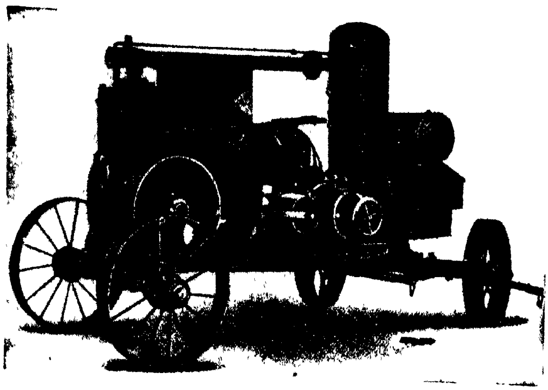


FIG. 16 Ingersoll Rand Portable Gasolene-Driven Compressor

service, the wheel being mounted directly on the crank-shaft, as in Figs. 19, 20. A plant somewhat similar in general layout was built by the Pelton Water Wheel Co., for the Alaska-Treadwell Gold Mining Co. Since the power developed is uniform throughout the revolution of the wheel, the compressor should be duplex to equalize the resistance as far as possible, and the rim of the wheel is made extra heavy, to act as a fly-wheel. Fig. 21 shows a recent form of Pelton wheel, sectionalized for mule-back transport in Mexico, the hub being designed for shrinking on forged bands, after the wheel is in place on its shaft. The fly-wheel effect of small diameter water-wheels is

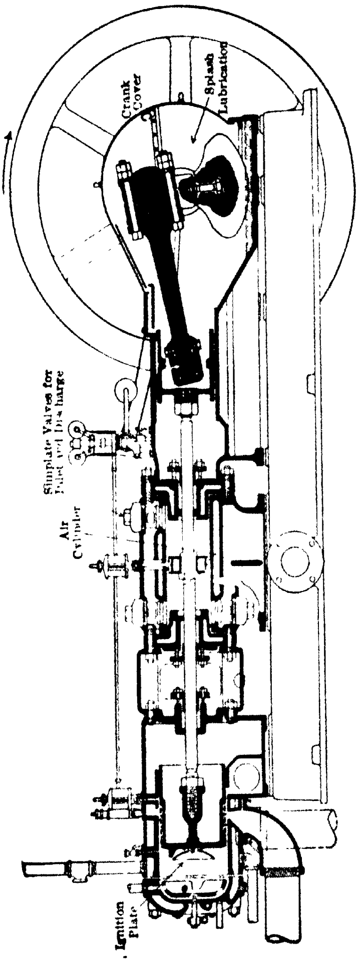


FIG. 17.—Straight-Line, Gasolene-Driven Compressor (Chicago Pneumatic Tool Co.)

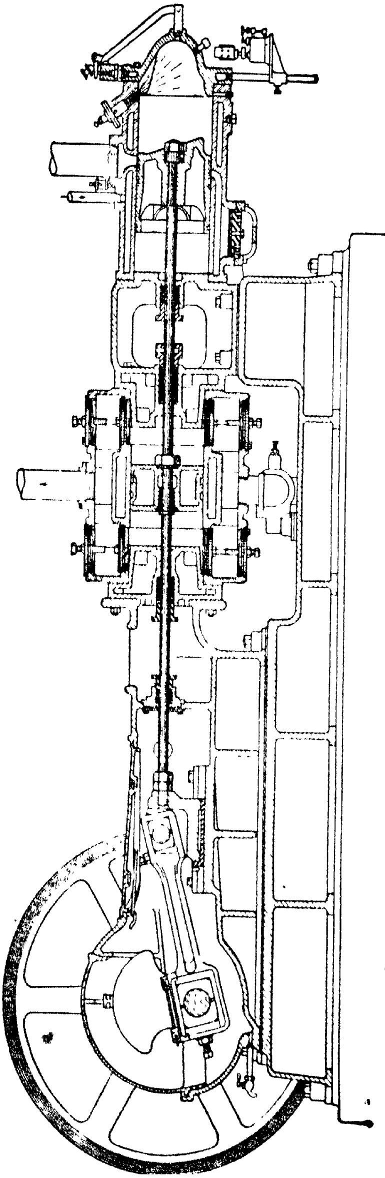


FIG. 18.—Straight-Line, Gasolene-Driven Compressor. Class "EO-1" (Ingersoll-Rand Co.)

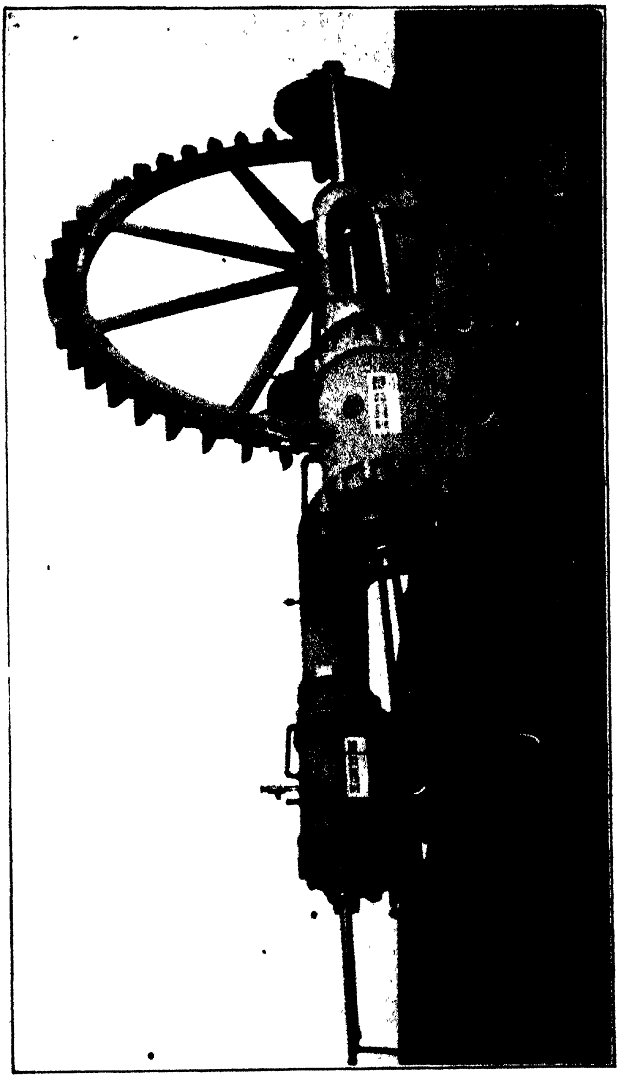


Fig. 19.—Duplex Compressor, with 16"×30" Cylinders, Direct-Connected to a 16-ft. Risdon Water-wheel, Working under a 300-ft. Head. Built by Risdon Iron Works Co. for the Goleta Mining Co.

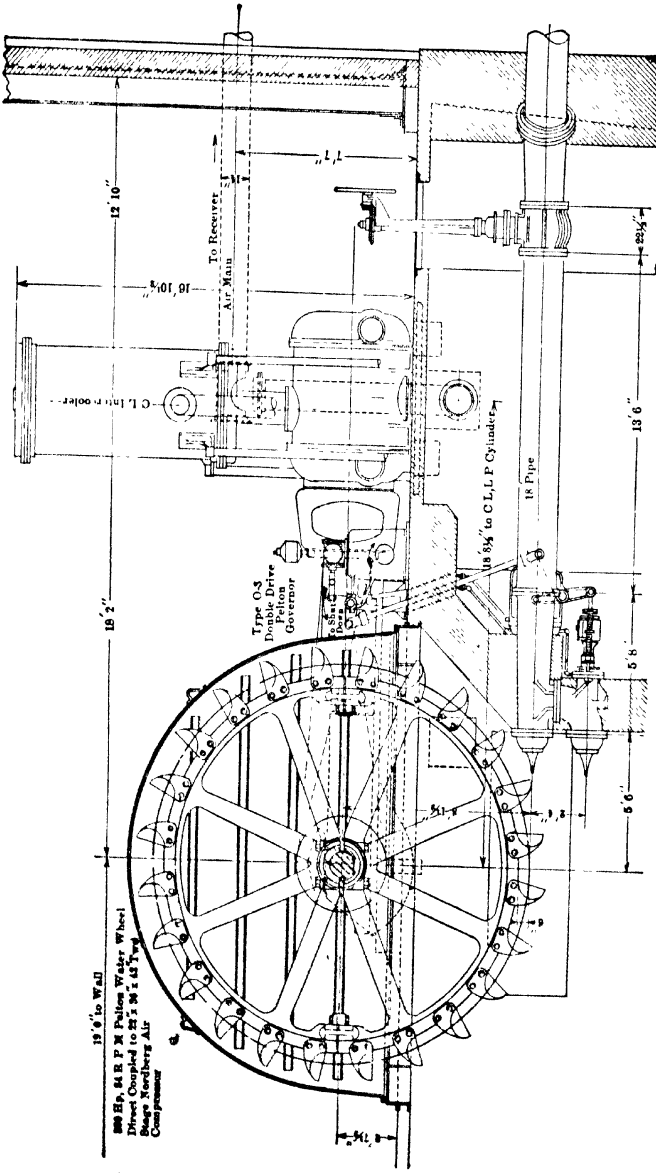


FIG. 20.—Nordberg Compressor, Direct-Connected to a Pelton Wheel

sometimes increased by heavy iron segments bolted on the spider. Water-driven compressors are also built by the Compressed Air Machinery Co., and others.

For best efficiency, the peripheral velocity of an impulse wheel is theoretically one-half the velocity of the jet of water from the nozzle. Hence, high heads involve high peripheral velocities, and for a wheel of small diameter a belt-drive would

FIG. 21.—Pelton Impulse Wheel.

be required. But belting or gearing can generally be avoided, except for a turbine-wheel drive (loss of power in belt transmission is, say, 8–10%). An impulse wheel can generally be made of large enough diameter to run at a peripheral speed that will insure economical use of the water, while still having a sufficiently low rotative speed for direct-connected compressors. To accomplish this under very heavy heads, the wheels are sometimes of great size.

At the Morning Mine, near Mullan, Idaho, is a large water-driven two-stage compressor. There are 4 cylinders, a high- and a low-pressure being set tandem on each side of a set of 3 Pelton wheels, mounted on the crank-shaft. A large volume of water, under a head of 140 ft., is delivered through 8,000 ft. of flume and 400 ft. of pressure pipe, driving two 12-ft. wheels. Two other streams, piped respectively $1\frac{1}{2}$ and 1 mile, produce heads of 1,140 and 1,420 ft.; these drive a 33-ft. Pelton wheel, placed on the crank-shaft between the smaller wheels. The central wheel is driven by separate jets from the high-pressure lines, and on account of their difference in head, the diameter of this wheel is a mean between the diameters theoretically necessary for obtaining a peripheral velocity properly proportioned to each head. The actual mean peripheral speed is 8,000 ft. per minute. To control the water under these great heads (pressures, about 490 and 610 lbs. per sq. in.), slow-acting gate valves are provided, with by-passes for starting and stopping. For stopping the compressor quickly the nozzles can be deflected clear of the wheel.

Each pair of cylinders are $33\frac{1}{2}$ and 18 ins. $\times$ 42-in. stroke; piston speed, 560 ft. The low-pressure cylinders compress to about 30 lbs., the high-pressure to 90 lbs. Inter- and after-coolers are placed in the tail-race of the smaller wheels. A positive valve-motion is employed for both inlet and discharge valves, which are of the Corliss type. On each side, parallel to the center line of the compressor and geared to the crank-shaft, is a long shaft, geared to which are short shafts carrying the valve eccentrics. As the discharge valves must open when the pistons are moving at nearly maximum velocity (800 ft. per min.), an auxiliary dash-pot allows them to open freely under the cylinder pressure, the positive eccentric motion closing them.

Indicator cards show this compressor to be highly efficient. An average of a number of cards gives mean pressures of: low-pressure cylinder, 17.86 lbs.; high-pressure, 41.14 lbs.; combined, 30.46 lbs. The mean theoretical adiabatic and isothermal pressures, corresponding to the combined mean are, respectively,

36.94 and 28.5 lbs. During the tests the observed temperatures were: cooling water, 38°; air at discharge from low-pressure cylinder, 135°; at high-pressure inlet, 46°; high-pressure discharge, 140°; on leaving the after-cooler, 62°. Mean atmospheric temperature, 55° and of the cooling water 38°.

If there is a sufficient volume of water, impulse wheels may be used with low heads, by introducing multiple nozzles, directed tangentially at two or more points of the periphery of the wheel. To prevent water from splashing over the compressor, the wheel is enclosed in a tight wooden or iron casing. The flow of water may be regulated by an ordinary gate valve; but if the head be great a special slow-moving gate must be used (as noted above), to avoid danger of rupturing the pressure pipe in case the compressor is suddenly stopped. Turbines are obviously not so well adapted for operating compressors as impulse wheels.

Nozzles are now usually of the "needle" type (Fig. 22). The position of the needle is adjusted by a hand-wheel and gearing for delivering the requisite volume of water, the needle being shaped to form a solid stream against the buckets of the wheel. The cut shows also an automatic cutoff governor, controlled by air pressure from the compressor receiver. An excess of receiver pressure, admitted to the small cylinder *a*, causes arm *b* to move backward, thus raising hood *c* in front of the nozzle and deflecting the water to the tail race, instead of allowing it to impinge on the wheel. When receiver pressure returns to normal, the hood automatically moves downward, out of line of the jet. This regulates the speed of the wheel (and also of the compressor), but does not economize water consumption by checking the nozzle flow. To save water, there are several devices for automatically regulating the opening of a valve in the pipe from penstock to nozzle, and so adjusting the stream impinging on the wheel.

In another design, the nozzle itself is deflected out of line with the buckets on the wheel.

Belt-driven Compressors. The fly-wheel is replaced by a belt-wheel with a heavy rim to give it sufficient weight. Figs. 23, 24 show straight-line and duplex, two-stage compressors,

Fig. 25 is a compact form of small two-stage compressor, suitable for underground installation, or where floor-space is limited. The Sullivan Machinery Co. also builds small belt-driven, single-stage compressors for surface and underground use, and a portable gasolene-driven compressor. For underground service, the " W K " and " W K-2 " types are mounted

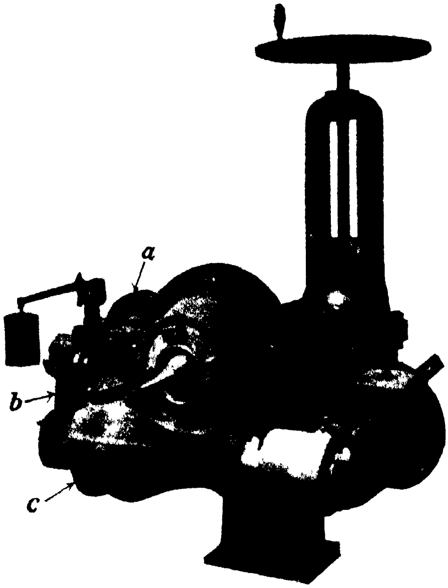


FIG. 22.—Needle Nozzle for Pelton Wheel

on a light, self-propelling truck; they are made in 4 sizes, capacity 98-258 cu.ft. of free air per min.

Fig. 26 shows a well-known English compressor made in a number of sizes for both steam and belt-drive. The two-stage machine has a differential piston, working in a single cylinder. Air is admitted to both the low- and high-pressure parts of the cylinder by one large piston valve, which controls absolutely the periods of admission and also the end of the discharge

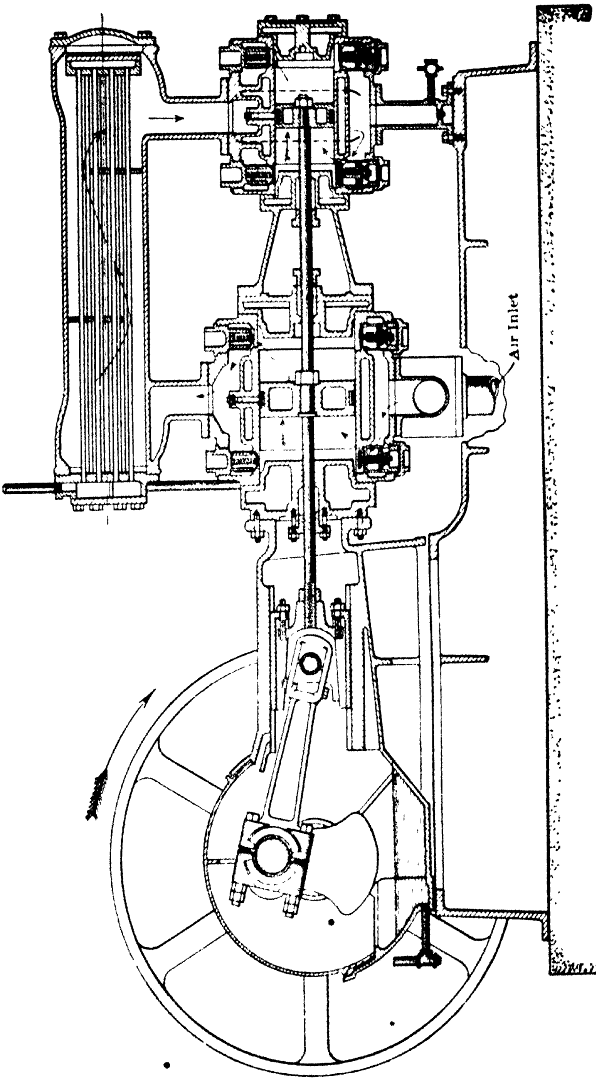
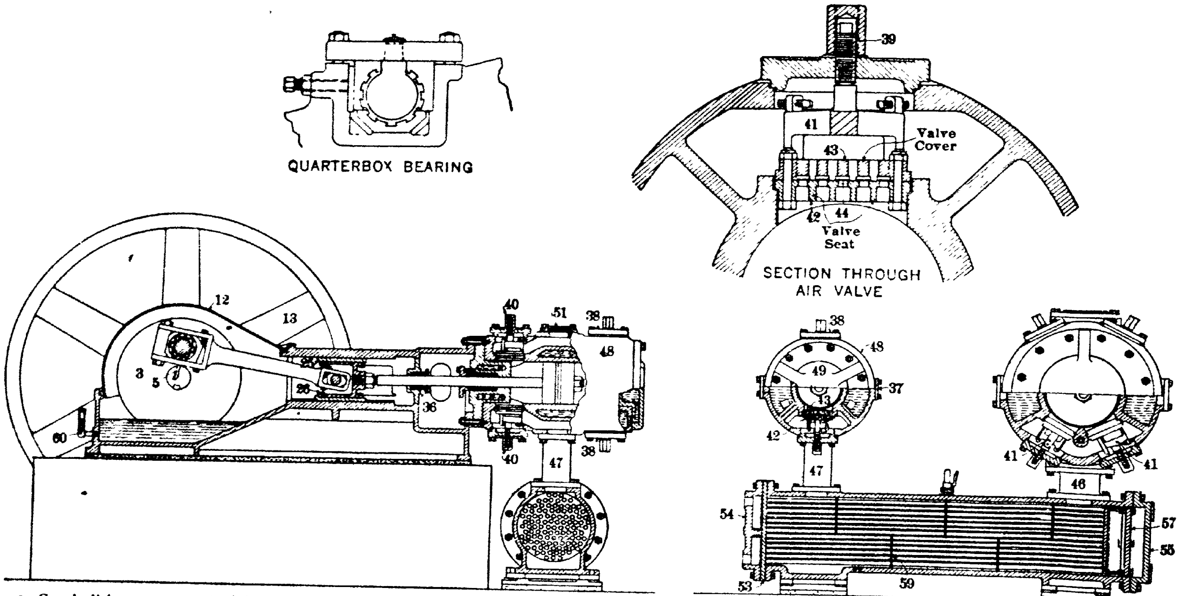


FIG. 23.—Sullivan Straight-Line, Two-Stage, Belt-Driven Compressor, Class "WH-3," 12" and 7¼"×10" Cylinders.



3. Crank disk.
5. Oil scoop.
12. Oil guard.
25, 26. Cross-head shoe and cap.
29. Low-pressure cyl.
36. Oil head and stuffer.
37. Hand-hole plate.
38. Valve cover.
39, 40. Valve screw and cap.
41. Valve yoke.
42. Valve seat.
43. Valve guard.
44. Valve strip (see Figs. 62, 63 Chap. VII).
45. Ream bolts and nuts.
46. Low-pressure disch pipe.
47. High-pressure intake pipe.
48. High-pressure cylinder.
50. Low-pressure suction flange.
51. High-pressure disch. flange.
54, 55. Front and rear cooler h'ds.
57. Floating head cover.
59. Baffle plates.
60. Oil gage.
62. Pop valve.

FIG. 24.—Laidlaw-Dunn-Gordon Duplex, Two-Stage, Belt-Driven Compressor, 14″ and 8½″×12″ Cylinders, Feather Air Valves.

periods. The actual discharge is through groups of poppet valves.

Fig. 27 is the high-pressure side of a recent duplex, two-stage compressor. Power may be derived from an engine already installed for other purposes, or from a water-wheel,

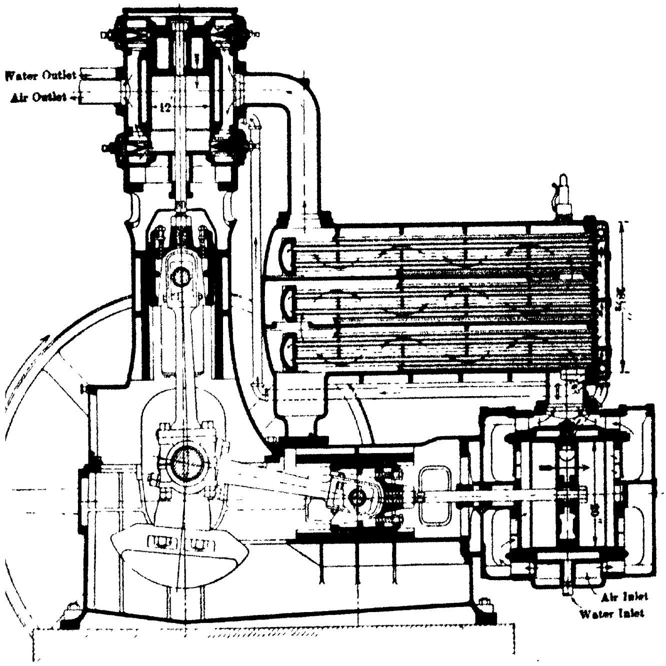


FIG. 25.—Sullivan Belt-Driven, Two-Stage, Angle Compressor, Class "WJ-3" (1917).

electric motor, or gasolene engine. Small portable motor-driven compressors mounted on wheels (like Fig. 16) are also to be had.

Some builders employ a "silent-chain" drive, when it is desired to place the motor close to the compressor and on the

same bed-frame, and at the same time to avoid the use of gearing (Fig. 28). It has high efficiency (about 95%), and will transmit up to, say, 200 H.P.

Compressor Direct-connected to Electric Motor. Although a belt-drive is preferable to gearing, at least for a compressor erected on the surface, geared electric-driven sets are sometimes

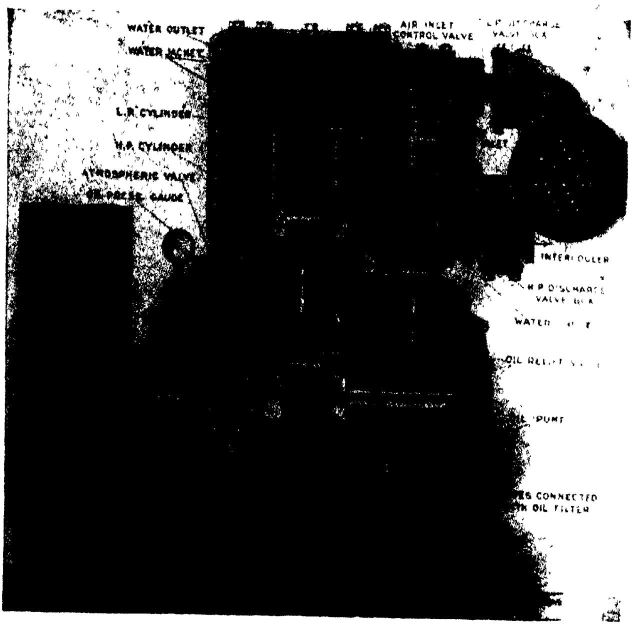


FIG. 26.—Alley and MacClellan Vertical, Two-Stage, Belt-Driven Compressor (from paper by G. Blake Walker, *Trans.*, Midland Inst., Min., Civil and Mech. Engs., Jan. 21, 1913).

used, a spur-gear on the crank-shaft engaging with a pinion on the armature. This design has been adopted for large plants, as, for example, at a two-stage installation of the Compañia de Peñoles, Mexico. By giving sufficient diameter and weight to the spur-wheel, it not only produces a low piston speed but serves also as a fly-wheel. Rawhide pinions are desirable to

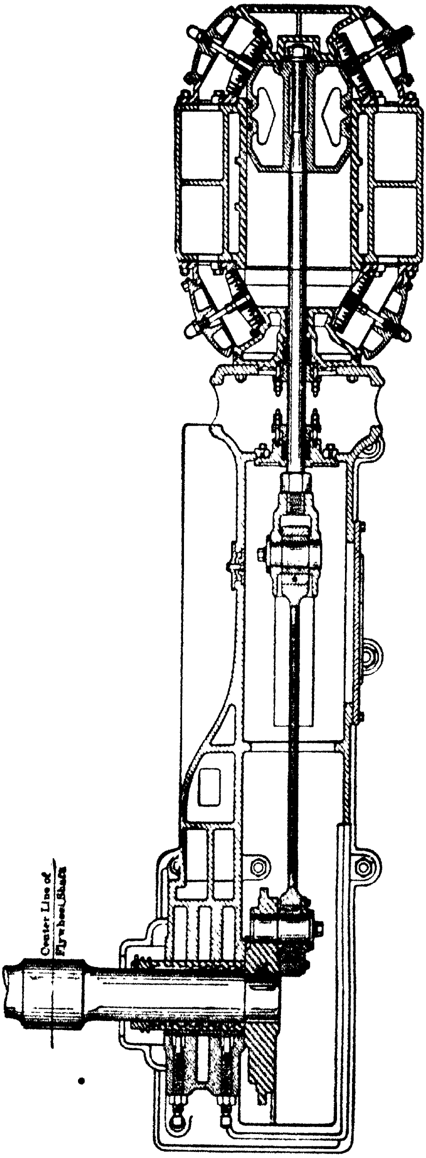


FIG. 27.—Ingersoll-Rand Duplex, Two-Stage, Belt-Driven Compressor, with Plate Valves. (High-pressure side shown.)

reduce noise. The small, high-speed Christensen compressor is well adapted for gearing to a motor, thus forming a compact machine where lightness or portability is essential. Fig. 29 shows a direct-connected, duplex compressor. The frame is inclosed and all parts are self-oiling, except the piston and cylinder. The crank-pit contains the oil supply. The oil is picked up by the edge of the crank disk, and taken off at the top

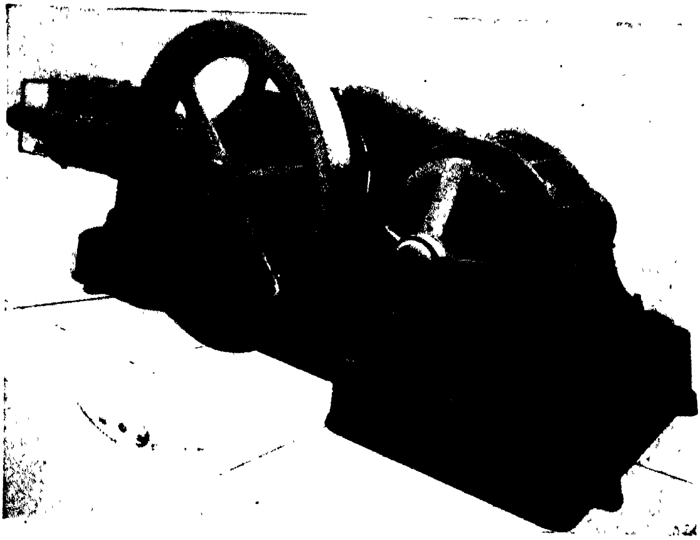


FIG. 28.—Ingersoll-Rand Compressor, Chain-Driven from an Electric Motor. Class "NE-1," 12¼"×12" Cylinder.

by a scraper; part goes to a distributing tank, from which small pipes lead to the crosshead and guides. The main bearing and crank-pin are oiled direct from the scraper by a projecting trough on each side. For the bearing the oil is led to a channel in the cap, and thence through a series of holes drilled in the bearing liner. Some of this oil goes to the collar of the eccentric hub, from which it is carried to the face of the eccentric through two holes. As the eccentric is closed, the surplus oil returns to the crank-pit. Thus the oil-feed is proportioned to the speed

of the compressor, ceasing entirely when the compressor is stopped, and results in a material saving.*

Fig. 30 is a well-known English direct-connected compressor. It has plate air valves (Chap. VII), forced lubrication, and large water-jackets.

These compressors are driven by direct-current, induction or synchronous motors, the rotors of which are of large diameter,

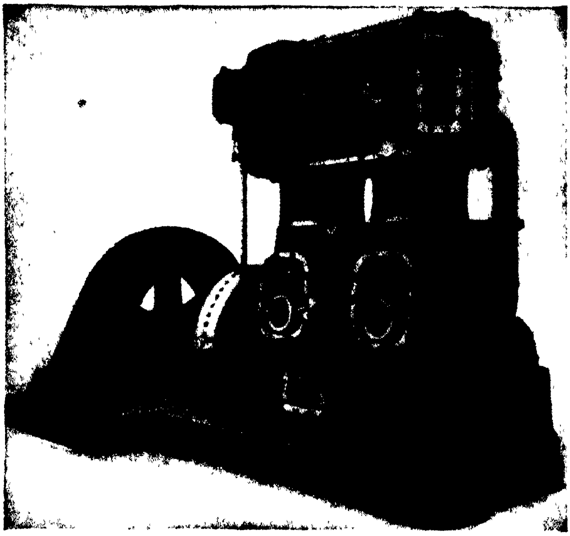


FIG 30.—Robey & Co. High-Speed, Vertical, Two-Stage, Motor-Driven Compressor

as shown, to produce a proper relation between peripheral and rotative speeds. Induction motors are good, as they run economically under wide variations of load.

Under proper conditions an electric-driven compressor may be erected underground, near the point of application of the air power. A difficulty in operating compressors underground

* Similar self-oiling ("flood and splash") systems are used for compressors shown in Figs. 1, 4, 9, 17, 18, 23, and 24

is in obtaining cooling water of good quality and in sufficient quantity. Mine water usually contains enough sediment to foul the inner surfaces of the jackets and intercooler. Also, the heated water must be cooled before reuse; for example, by pumping it through a worm pipe laid in the sump. Hence, the volume of water entering and pumped from the sump is important. While a small compressor (of say 300 cu.ft. free air per min.) might be successful underground, a large one would be out of the question.

Turbo-compressors. Based on the principle of his steam turbine, Parsons designed a type of low-pressure compressor, useful as a blowing engine. From this, Rateau developed the turbo-compressor, now built by the Westinghouse Electric Co., Ingersoll-Rand Co., and other makers in Europe and the United States. They satisfactorily produce pressures suitable for mine service. For pressures above 2 or 3 atmospheres, the compression is done in stages, the number of impellers per stage depending on the required pressure. Fig. 31 shows a 2-stage turbo, with 10 impellers per stage.

The rapid rotation of the impellers (rarely less than 3,000 R.P.M.) imparts to the air a velocity of 300 ft. or more per second. The velocity at which the air issues from each impeller is converted into head, or pressure, as the air passes into the larger passages leading to the next impeller, each adding an increment of pressure. This increment is normally 2–2½ lbs., sometimes 3–4 lbs., or even more.

Water jackets are applied to the spaces between the impeller casings, diffusers and passages. In the first impellers of the series, where the pressure is low, the temperature rises rapidly, notwithstanding the jacketing; but, as the density of the air increases, cooling becomes more effective, and the total isothermal efficiency is good. The volume of cooling water required, at 70° F., is roughly 165 gals. per min. per 1,000 H.P. Some engineers estimate 3,000 gals. per hr. per 1,000 cu.ft. free air compressed per min.

Examples. On the Rand, South Africa, a number of large turbos are operated by the Victoria Falls and Transvaal Power

Co. Power is transmitted from central stations through more than 18 miles of piping to 17 mines. The last compressor installed has a capacity of 58,800 cu.ft. free air per min. to 140–170 lbs. gage, requiring 12,000–13,000 H.P. It is driven by a steam turbine using superheated steam at 170 lbs.; speed, 3,000 R.P.M. There are 3 sets of impellers (3 stages). The same company has 12 4,000-H.P. turbos in operation, each delivering 20,000–23,000 cu.ft. free air per min., compressed to 135–170 lbs., and driven by synchronous motors. They are 4-stage: 2 low-pressure, 1 intermediate and 1 high-pressure cylinders; speed, 3,000 R.P.M. Efficiency (referred to isothermal compression), 67.5%.

Fig. 31 shows a 2-stage turbo, driven by a steam turbine; capacity, 7,500 cu.ft. per min. to 80 lbs.; brake H.P., 1,480, at 4,000 R.P.M. Each stage comprises 10 impellers, similar to those in Fig. 32 (for details, see *Trans.* Instn. Min. Engs., England, Vol. 45).

Another turbo, driven at 4,200 R.P.M. by a 1,000-H.P. mixed-pressure turbine, compresses 4,400 cu.ft. air per min. to 99 lbs. Consumption of exhaust steam (taken at 15.6 lbs. and discharged

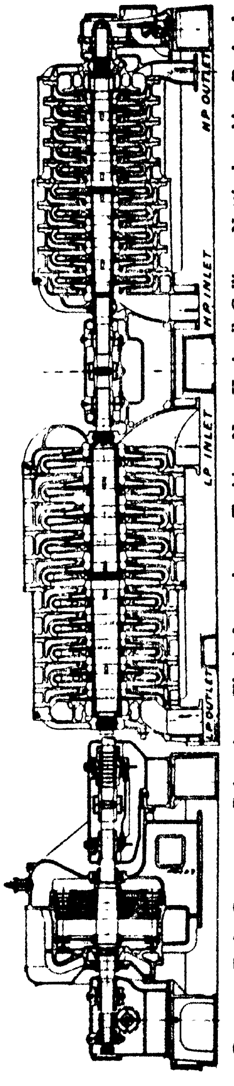


Fig. 31.—Turbo-Compressor, Driven by a Westinghouse Steam Turbine, New Hucknall Colliery, Nottinghamshire, England. (G. Blake Walker.)

at 1.14 lb.) was 7.98 lbs. per 100 cu.ft. free air; with this were used 5.05 lbs. live steam at 85 lbs.

The final temperature for a delivery pressure of 85–115 lbs. is say 130°–170° F., depending on temperature of the cooling water. In one case, with cooling water at 50° F., and a delivery pressure of 120 lbs., the final temperature was 136° F.

Turbos compressing 5,000–12,000 cu.ft. free air per min. to 85–120 lbs. should run about 3,800 R.P.M.; smaller machines, down to 2,500 cu.ft. at 70–85 lbs., 1,750 cu.ft. at 60 lbs., and 600 cu.ft. at 25 lbs., should be designed for 4,200–5,000 R.P.M. (G. Blake Walker, *Trans.* Instn. Min. Engs., England, Vol. 44, pp. 629–689).

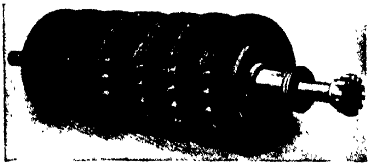


FIG. 32.—Series of Impellers, Westinghouse Turbo-Compressor.

Fig. 33 shows results of tests on the turbo at the New Hucknall colliery, England. Turbos exceeding 6,000 cu.ft. free air capacity should give 70–80% efficiency (referred to isothermal); between 3,000 and 6,000 cu.ft., 65–70% efficiency at full load, and under 50% at less than half load.

Field of Use. An important use for turbo-compressors is in furnishing blast for smelting furnaces, Bessemer converters, and other metallurgical service; air pressure being, say, 5–35 or 40 lbs. The chief difference between blowers and high-pressure turbos is in the number of impellers and provision for dealing with the heat of compression. The efficient governors of some recent turbos give a nearly constant pressure, even with quite wide fluctuations in the volume of air used. For example, a 5,000-cu.ft. Thomson-Houston turbo, running normally at 4,600 R.P.M., showed a variation of only 2 or 3 lbs. for a range in output from 1,500 to 6,000 cu.ft. free air per min.

In point of first cost, it is generally not advisable to use turbos for capacities smaller than 2,200–2,500 cu.ft. free air per min., nor for pressures exceeding 70–85 lbs.; for smaller output or higher pressure, reciprocating compressors are best.

NOTE.—It is not practicable in a book that is not a trade publication, to describe all the makes of air compressors. Those which have been instanced illustrate the chief features of design. The same is true regarding the descriptions of air valves, etc., in Chaps. VII, VIII, and IX. It must not be understood that the compressors referred to are considered the only good ones, nor that the author by omitting to mention and insert cuts of all compressors, desires to discriminate against those that are less well known. Most of the European compressors, including many excellent machines, are omitted altogether, though references to the valve motions of some of them will be found under the appropriate heads.

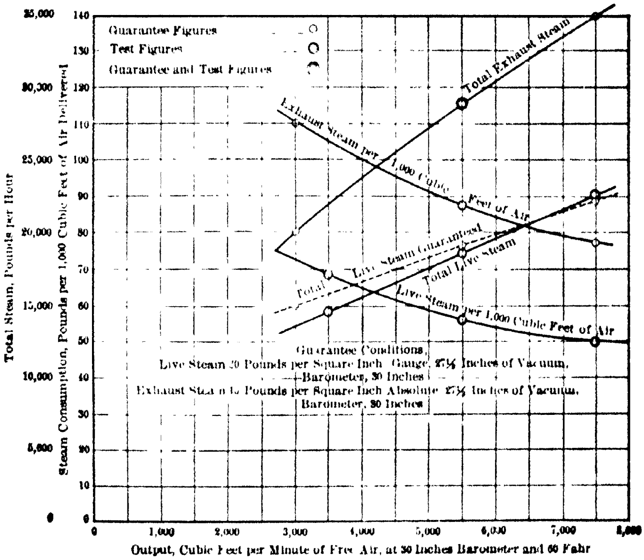


FIG. 33.—Results of Test of British-Westinghouse Rateau Mixed-Pressure Turbine and Air Compressor. Speed, 4,000 R.P.M.; output, 7,500 cu.ft. free air per min., at 80 lbs. gage. Test made by W. F. Mylan (*Trans.* Instn. Min. Engs., England, Vol. 45, pp. 245–264).

Following is a detailed classification of the compressor practice of the Ingersoll-Rand Co. Attention is called to the note regarding the two-stage, straight-line compressors of this make. Several other builders (for example, the Sullivan Machinery Co. and the Norwalk Iron Works Co.) furnish two-

stage, straight-line compressors for the ordinary pressures used for rock-drills, pumps, etc.

STEAM DRIVEN

Straight-line.	Single-stage, double acting.	Direct-connected to steam engine. For long belting to steam engine. For short belting to steam engine
	NOTE. Two-stage, straight-line compressors for steam operation are built for high-pressure work only	
	Single-stage, double acting, portable.	Direct-connected to steam engine.
Duplex, small and medium capacity.	Single-stage. Two-stage.	Direct-connected to steam engine. For long belting to steam engine. For short belting to steam engine.
Duplex, Large large capacity.	Single-stage. Two-stage.	Direct-connected to steam engine
Turbo-compressors.		Direct-connected to steam turbine

POWER DRIVEN

Vertical.	Single-stage, single-acting, stationary	For long belting to line shaft, gasolene engine or electric motor. For short belt drive.
	Duplex single-stage, single-acting, portable.	For short belt drive.
Straight-line.	Single-stage, double-acting.	For long belt drive. For short belt drive. For chain drive.
	NOTE.—Two-stage compressors of this type are built for high-pressure work only.	
	Single-stage, double-acting, portable.	For short belt drive. For chain drive.
Duplex, small and medium capacity	Single-stage Two-stage	For long belt drive. For short belt drive.
Duplex, large capacity.	Single-stage. Two-stage.	For long belt drive. Direct-connected to electric motor, gasolene, gas, or oil engine, or water wheel.

The following alphabetical list, while incomplete, comprises the names of most of the American compressor-builders.

Allis-Chalmers Manufacturing Co.
American Air Compressor Works
Bury Compressor Co.
Chicago Pneumatic Tool Co
Clayton Air Compressor Works.
Compressed Air Machinery Co.
Ingersoll-Rand Co.
Laidlaw-Dunn-Gordon Co.
New York Air Compressor Co.
Nordberg Manufacturing Co.
Norwalk Iron Works Co.
Rix Compressor and Drill Co.
Sullivan Machinery Co.
Vulcan Iron Works.
Worthington Pump and Machine Co.

CHAPTER III

THE COMPRESSION OF AIR

In the production and use of compressed air occur serious losses, which to a large extent are unavoidable. Even in the best compressors the efficiency, or ratio of the force stored up in the compressed air to the work expended in compressing it, rarely exceeds 75% and often falls below 60%. To understand the causes of these losses it is necessary to study the principles of air compression. This study is advisable, also, before proceeding to a description of the air end of the compressor. Several definitions follow:

"**Free air**" is air at normal atmospheric pressure, as taken into the compressor cylinder. But since atmospheric pressure varies with the altitude above sea-level, and with the barometric reading at any particular time or place, the expression "free air" has no precise signification, with respect to the pressure, volume, and temperature of the air. At sea-level it is in reality "compressed air," at the normal atmospheric pressure of 14.7 lbs. per sq. in. As commonly employed the term means air at sea-level pressure, and at a temperature of 60° F.

The **absolute pressure** of air is measured from zero, and is equal to the assumed (or observed) atmospheric pressure plus gage pressure; ordinary gages register pressures in lbs. per sq. in. above atmospheric pressure.

Absolute temperature is the temperature as measured from the "absolute zero" point, which is 491.4° F. below the freezing-point of water, or say 459° below zero F. Thus 60° F. of thermometric temperature is equivalent to an absolute temperature of $459° + 60° = 519°$ F.

Two fundamental laws govern the behavior of a perfect gas when undergoing compression, which for practical purposes

apply also to atmospheric air. In discussing the problems of air compression, all the relations between volume, pressure, and temperature may be expressed in accordance with these laws. The first law (Boyle's) is: At constant temperature the volume occupied by a given weight of air varies inversely as the pressure. This is expressed by:

$$PV = P'V' = \text{constant}; \text{ or } \frac{P'}{P} = \frac{V}{V'}; \text{ in which}$$

V = volume of the given weight of air (or gas) at the freezing-point and at a pressure P (V usually being taken as the volume in cu.ft. occupied by 1 lb. of air); V' = volume of the same weight of air at the same temperature and at any pressure P' (the pressures being absolute pressures).

For example, to compress a quantity of atmospheric air at constant temperature to 0.147 of its original volume (atmospheric pressure being 14.7 lbs.), requires a pressure of 100 lbs. per sq. in.; when compressed to 0.074 of its original volume, the pressure required is 200 lbs., and so on.

TABLE I (D. K. CLARK)

Temperature, Deg. F.	Weight of 1 Cu.ft. in Lbs.	Volume of 1 Lb. in Cu.ft.	Temperature, Deg. F.	Weight of 1 Cu.ft. in Lbs.	Volume of 1 Lb. in Cu.ft.
0	0863	11 582	110	0697	14 345
10	0845	11 834	120	0685	14 596
20	0827	12 085	130	0674	14 847
30	0811	12 336	140	0662	15 098
32	0807	12 386	150	0651	15 350
40	0794	12 587	160	0641	15 601
50	0779	12 838	170	0631	15 852
60	0764	13 089	180	0621	16 103
62	0761	13 141	190	0612	16 354
70	0750	13 340	200	0602	16 605
80	0736	13 592	210	0593	16 856
90	0722	13 843	212	.0591	16 907
100	0710	14 094			

Table I shows the weight and volume of dry air, at temperatures from 0°–212° F., and at atmospheric pressure.

The production and use of compressed air are not governed solely by Boyle's law. During compression heat is generated, and when the air is allowed to re-expand to its original volume this heat is taken up. If there is no transference of heat, the internal work, manifested by the increase of temperature, is independent of the time occupied by compression. This condition is expressed by the second law, that of Charles and Gay-Lussac: When under constant pressure, the volume of a gas expands or contracts for each degree rise or fall of temperature, from freezing to boiling, by a constant fraction of the volume which is occupied at the freezing-point. Stated in another way, the volume of a gas under constant pressure is nearly proportional to the absolute temperature. The equation may be written: $V' = V(1 + at^{\circ})$. The complete relations between pressure, volume, and temperature are expressed by the equation: $P'V' = P'V(1 + at^{\circ})$, in which P' and V' represent the pressure and volume of a given weight of air (or gas) at t° F. above the freezing-point, V is the volume of the same quantity at the freezing-point, and a the coefficient of expansion of air, which is practically constant and is very nearly $\frac{1}{491}$ on the Fahrenheit scale Hence, for a rise in temperature of 1° F., the volume of air increases by $\frac{1}{491}$ of the volume occupied at the freezing-point, under the same pressure (491° F. being the absolute temperature below freezing).

The practical application of this law is that the heat generated reacts upon the air under compression, and increases the pressure due merely to the reduction in volume. By cooling the compressed air to its original temperature the pressure would be reduced to the normal amount, according to the first law. That is, the heat produced by compressing a given volume of air corresponds in degree to the cold resulting from the re-expansion of the same quantity of air to its original volume and pressure.

Two other statements may be deduced from what precedes: 1. Under constant pressure the volume of air varies directly as the absolute temperature; 2. The volume being constant, the absolute pressure varies directly as the absolute temperature,

The first of these statements is expressed thus:

$$\frac{V}{t}=\frac{V'}{t'}=\text{constant},$$

in which t and t' are absolute temperatures; whence, from Boyle's law:

$$P\frac{V}{t}=P'\frac{V'}{t'}=\text{constant}.$$

For convenience, this constant is commonly denoted by R, and the general equation is written $PV=Rt$, or $\frac{PV}{t}=R$.

The value of R is found as follows: Since for a given weight of gas or air the density, D, is inversely proportionate to the volume, $V=\frac{1}{.08073}$, .08073 being the weight in lbs. of 1 cu.ft. of dry air, at sea-level pressure (14.7 lbs.) and 32° F. The normal atmospheric pressure per sq. ft $=14.7\times 144=2,116.8$ lbs If, therefore, 1 cu.ft. be expanded by the application of heat to a volume of 2 cu.ft., the work done against atmospheric pressure, per lb. of air, will be $\frac{2116.8\times 1}{.08073}=26,220$ ft. lbs. To double the volume, according to Charles' law, would require the expenditure of 491.4° F. of heat; whence, in raising the temperature 1° F., the external work done by expansion is:

$$\frac{PV}{t}=\frac{26220}{491.4}=53.37=R.$$

The heat generated during compression and corresponding to different pressures is shown in Table II, the volume at normal atmospheric pressure being 1, at a temperature of 60° F. This table shows that the *rate* of increase of temperature is not uniform, but diminishes as the pressure rises. Thus, from 1 to 2 atmospheres the increase is 115.8°; from 2 to 3, 79.3°; from 3 to 4 atmospheres, 62.3°, etc. The quantity of heat

generated during compression may be calculated by the following formula:

$$Q=\frac{R\times t}{J}\times \text{Nap. log.}\frac{V'}{V}, \text{ in which}$$

Q = quantity of heat in thermal units;

R = constant = 96.037 (French unit) or 53.37 (English unit);

t = absolute final temperature in degrees, corresponding to V' (centigrade scale for French and Fahrenheit for English units);

J = value of one thermal unit = 1,400 ft. lbs. (or 778 ft. lbs. if English units be used);

V and V' = volumes of air, cu.ft., at beginning and end of compression.

TABLE II

Pressure in Atmospheres	Absolute Pressures, Lbs. per Sq. in. above Vacuum	Volumes in Cu. ft. Adiabatic Compression	Final Temperatures Deg. F.	Corresponding Increases of Temperature.
1.00	14.70	1.000	60.0	00.0
1.25	18.37	0.854	94.8	34.8
1.50	22.05	0.750	124.9	64.9
2.00	29.40	0.612	175.8	115.8
2.50	36.70	0.522	218.3	158.3
3.00	44.10	0.459	255.1	195.1
3.50	51.40	0.411	287.8	227.8
4.00	58.80	0.374	317.4	257.4
5.00	73.50	0.319	369.4	309.4
6.00	88.20	0.281	414.5	354.5
7.00	102.90	0.252	454.5	394.5
8.00	117.60	0.229	490.6	430.6
9.00	132.30	0.211	523.7	463.4
10.00	147.00	0.195	554.0	494.0
15.00	220.50	0.147	681.0	621.0

As the rise in temperature due to compression is proportional to the ratio of the final absolute pressure to the initial absolute pressure, the quantity of heat generated during compression to any given pressure, and the consequent work done is greater at high altitudes than at sea-level.

The above conclusions are illustrated by the diagram, Fig. 34.* It is, in reality, two diagrams, combined to save space. *First*, beginning at the lower left-hand corner, and curving up-

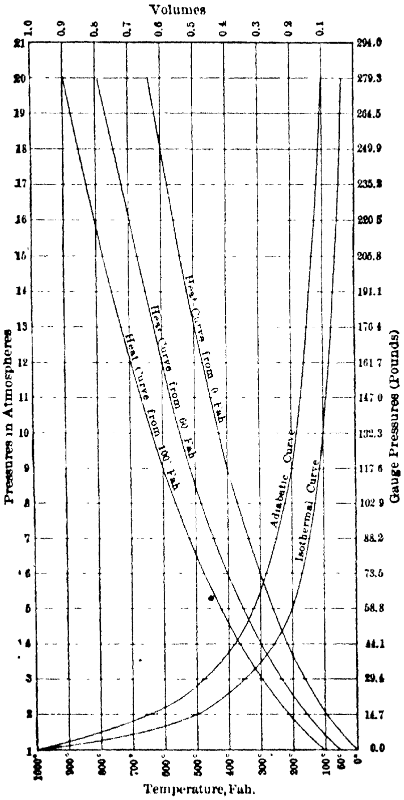


FIG. 34.

ward, are the adiabatic and isothermal compression lines. Their intersections with the horizontal and vertical lines give the volumes of the unit of air when subjected to any given

* Taken from "Compressed Air Production," by W. L. Saunders, several slight corrections having been made in the adiabatic and isothermal lines.

pressure, by reading the figures at the top, and right- or left-hand margin of the diagram. The initial volume is taken as 1, and the spaces between the vertical lines are each one-tenth. The resulting volume is independent of the initial temperature of the air. The corresponding pressure may be read in terms of either gage or atmospheric pressure. *Second*, beginning at the lower right-hand corner of the diagram, and rising toward the left, are the lines of temperature, the assumed initial temperatures being 0°, 60° and 100° F. The temperature corresponding to any given pressure is read on the lower margin. It should be observed that these temperature curves are those of adiabatic compression.

It follows from the above that if the temperature of the air rises during compression an increase of work ensues.

Isothermal and Adiabatic Compression. In accordance with the laws already stated, air may be compressed in two ways:

Isothermal Compression. The temperature is kept constant during compression, the heat generated being abstracted as fast as it is produced. In this case the pressure of the air varies according to the equation $PV = P'V'$, and the compression curve of an indicator diagram is an isothermal curve.

Adiabatic Compression. The temperature may be allowed to rise unchecked during the period of compression, as it will when there is no transference of heat by radiation or cooling devices. The rise in temperature increases the pressure due to reduction of volume only. Thus, the pressure rises faster than the volume diminishes, and $\frac{P'}{P}$ becomes greater than $\frac{V}{V'}$.

This relation is determined by considering the specific heats of air at constant pressure and at constant volume. The specific heat of any gas or vapor at constant pressure, C_p, is the quantity of heat required to raise the temperature of 1 lb. of the gas 1° F., the pressure being unchanged. The specific heat at constant volume, C_v, is the quantity of heat required to raise the temperature of the gas 1° F., the volume being unchanged. Regnault found that for air $C_p = 0.2375$ and $C_v = 0.1689$. C_p is the greater, because external work is done

during a change of temperature, if the pressure be constant and the air free to expand; under constant volume, no work is done upon external resistances. When, as in adiabatic compression the heat generated reacts on the air under compression and increases the value of $\frac{P'}{P}$, to maintain the equation $\frac{V}{V'}$ must be increased by an amount equivalent to the external work performed. The specific heats may be expressed in heat units, as above; or, by multiplying them by the mechanical equivalent of a heat unit (778 ft.-lbs. = J), they are given in terms of ft.-lbs. and are then denoted by K; that is,

$$JC_p = K_p \text{ and } JC_v = K_v.$$

Since $K_p = C_p \times 778 = 184.77$, and $K_v = C_v \times 778 = 131.4$:

$$\frac{K_p}{K_v} = \frac{184.77}{131.4} = 1.406.$$

This ratio is commonly denoted by n, and is the exponent of the power to which $\frac{V}{V'}$ must be raised to make it equal to $\frac{P'}{P}$*; n may also be expressed as equal to the ratio of the specific heats at constant pressure and volume:

$$\frac{C_p}{C_v} = \frac{0.2375}{0.1689} = 1.406 = n.$$

The general equation for adiabatic compression is therefore:

$$PV^n = P'V'^n \text{ or } \frac{P'}{P} = \left(\frac{V}{V'}\right)^{n=1.406}$$

Work of Compressors without Clearance. Isothermal Compression. The work done by a compressor without clearance, and using isothermal compression, is represented by the area

* A statement of the proof of this deduction is unnecessary here; it is given in several books on Thermodynamics, for example, in Perry's work on the Steam Engine, p. 333.

under the compression curve (Fig. 35). **Let AB be an isothermal** curve, AD representing any volume V of free air, and BC the volume V′, to which this quantity of air is compressed; the corresponding absolute pressures being respectively P and P′. The curve is an equilateral hyperbola, and the work W (represented by the area ABCD) $= W_1 + W_2 - W_3$, in which

W_1 = area under AB $= \int PdV$.

W_2 = area under BC $= P'V'$, representing the work of expelling the air from the cylinder.

W_3 = area under DA $= PV$, representing the negative work done by atmospheric pressure on the suction or intake side of the piston.

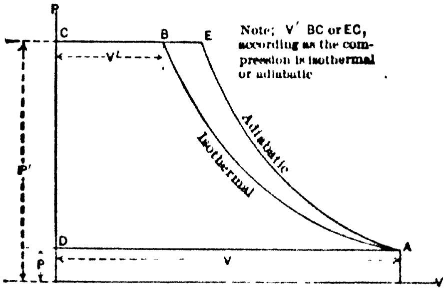


FIG. 35.—Reference Diagram.

Since $PV = P'V'$, W_2 and W_3 cancel, so that the algebraic sum of

$$W = W_1 + W_2 - W_3 = \int PdV \quad . \quad . \quad . \quad . \quad . \quad (1)$$

To integrate this expression, substitute for P its equivalent $\frac{P'V'}{V}$:

$$W = \int_{V'}^{V} P'V' \frac{dV}{V} = P'V' \int_{V'}^{V} \frac{dV}{V}.$$

Integrating: $\quad W = P'V' \times \text{Nap. log.} \left(\frac{V}{V'}\right)^{*} \quad . \quad . \quad . \quad . \quad . \quad (2)$

* The Naperian or hyperbolic logarithm of a number, generally written "$\log_e$," is obtained by multiplying the *common* logarithm by the constant 2.302585

The equation may also be written:

$$W = PV \log_e\left(\frac{P'}{P}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

a form convenient for use in making air compressor calculations. When expressed in ft.-lbs. (by putting V in terms of cu.ft., and P,P′ in lbs. per sq. in.):

$$W = 144\, PV \log_e\left(\frac{P'}{P}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

which is the general equation for the work of compressors operating isothermally and without clearance.

Work of Compressors without Clearance. Adiabatic Compression. Referring to Fig. 35, the line AD represents the initial volume, V, of air at normal atmospheric pressure, and the line EC the final volume V′, to which the same quantity of air is compressed; that is, V′ is the volume after the compression part of the stroke is completed and before delivery begins. In undergoing this change of volume, the pressure increases from P to P′, and the resulting compression line AE is an adiabatic curve, following the law:

$$PV^n = P'V'^n = C \text{ (constant), or } P = \frac{C}{V^n} \quad . \quad . \quad . \quad (5)$$

The total work done in the compressing cylinder is:

$$W = (W_1 + W_2 - W_3), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

in which:

W_1 = the work of compression.

W_2 = work required to force the compressed air out of the cylinder, into the receiver.

W_3 = work done by atmospheric pressure on the suction side of the piston, while the inlet air is entering the cylinder.

First.—The work W_1, in ft.-lbs., is:

$$W_1 = \int_{V'}^{V} 144 P dV$$

Substituting the value of P now expressed in lbs. per sq.in., from equation (5):

$$W_1 = 144\int_{V'}^{V}\frac{CdV}{V^n}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Integrating between the limits V and V′:

$$W_1 = 144C\left[\frac{V^{(1-n)} - V'^{(1-n)}}{1-n}\right]. \quad . \quad . \quad . \quad . \quad (8)$$

Dividing the second member of the equation by (-1) and substituting for C its value PV^n:

$$W_1 = \frac{144PV^n}{n-1}\left[\frac{1}{V'^{(n-1)}} - \frac{1}{V^{(n-1)}}\right]. \quad . \quad . \quad . \quad . \quad (9)$$

$$= \frac{144PV}{n-1}\left[\frac{V^{(n-1)}}{V'^{(n-1)}} - 1\right]. \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

But, since $\frac{P'}{P} = \frac{V^n}{V'^n}$, $\frac{V}{V'} = \left(\frac{P'}{P}\right)^{\frac{1}{n}}$, which, raised to the $n-1$ power, gives:

$$\frac{V^{(n-1)}}{V'^{(n-1)}} = \left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Substituting this value in (10):

$$W_1 = \frac{144PV}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]. \quad . \quad . \quad . \quad . \quad (12)$$

Second.—The work W_2, of expelling the air from the cylinder,

$$= 144P'V' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Multiplying by $\frac{V'}{V}$ both members of the expression, $\frac{P'}{P} = \left(\frac{V}{V'}\right)^n$:

$$\frac{P'V'}{PV} = \left(\frac{V}{V'}\right)^{n-1}; \text{ whence, } P'V' = PV\left(\frac{V}{V'}\right)^{n-1}.$$

But,

$$\frac{V}{V'} = \left(\frac{P'}{P}\right)^{\frac{1}{n}} \text{ and } \left(\frac{V}{V'}\right)^{n-1} = \left(\frac{P'}{P}\right)^{\frac{n-1}{n}}; \text{ hence}$$

$P'V' = PV\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}$; which, substituted in equation (13), gives:

$$W_2 = 144P'V' = 144PV\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \quad . \quad . \quad . \quad . \quad (14)$$

Third.—The work W_3, done by atmospheric pressure on the back of the piston, $=144PV$ . . . . (15)

Taking the algebraic sum of W_1, W_2 and W_3, from equations *(12)*, *(14)* and *(15)*, and substituting in equation (6):

$$W=144\left\{\frac{PV}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}-1\right]+PV\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}-PV\right\};$$

whence, by reducing to a common denominator:

$$W=144\frac{PV\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}-1\right]+(n-1)PV\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}-(n-1)PV}{n-1}$$

and cancelling:

$$W=\frac{144PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}-1\right] \quad . \quad . \quad . \quad . \quad (16)$$

which is the general expression for the work of single-stage compressors, with adiabatic compression, and when clearance is zero.

The relations between the two conditions of compression are represented graphically by Fig. 36. By laying off to scale the

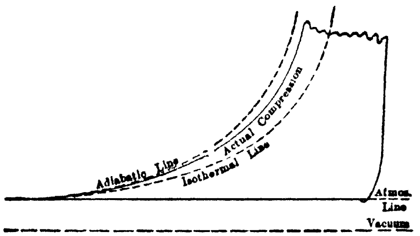


FIG. 36.

volumes of air on the horizontal line of the diagram, the corresponding pressures at different points of the stroke of the piston are measured on the verticals. The adiabatic curve rises more rapidly than the isothermal, meaning that more work is expended. Perfect isothermal compression is unattainable. It is only approximated even with the best cooling arrangements, and running the compressor at a very slow speed. On the other hand, if the air compressed adiabatically could be kept hot

until used, the loss of the additional work which was expended in compressing it would be prevented. But neither can this be done. The air is usually conveyed considerable distances before it is used, and radiation from the pipes soon reduces the pressure to that corresponding with the temperature of the surrounding atmosphere. In practice, a combination of the two theoretical modes of compression is employed, the net result depending upon the degree of perfection of the compressor and of the cooling arrangements. When compressing in a single cylinder to 60 or 80 lbs., and a piston speed not exceeding 300 ft. per min., it is probable that about one-half of the total possible cooling is all that may be expected.* The aim is to begin compression with the air at a low initial temperature, and to bring the compression line as close as possible to the isothermal line. The air must be cooled thoroughly during compression and before it leaves the cylinder; any subsequent cooling, in the receiver or in the air main, entails loss.

In ordinary practice the abstraction of heat during compression is very imperfect. Some distance must be traversed by the compressing piston before there is any considerable rise in temperature, and until the temperature does rise no cooling can be effected. The abstraction of heat does not begin at the beginning of the stroke. The temperatures of the intake air and of the cooling water are likely to be nearly the same, so that all the possible reduction of temperature in any one cylinderful of air must take place in a period of time less than that occupied in making the stroke. In modern dry compressors of fairly large size, and running at full working speed, the compression line is usually much nearer the adiabatic than the isothermal curve, and often follows the adiabatic curve quite closely.

The heat produced by compression may be absorbed:

1. By introducing cold water into the air cylinder.

2. By cooling the cylinder from without, enveloping it in a cold-water jacket.

Machines of the first class are known as "wet compressors"; those of the second, "dry compressors."

* Frank Richards, "Compressed Air," p. 66.

Values of *n* in the equation $\frac{P'}{P} = \left(\frac{V}{V'}\right)^n$. In purely adiabatic compression, $n = 1.406$; in ordinary single-cylinder dry compressors, n is roughly 1.3, while in the best single-stage wet compressors (with spray injection) n becomes 1.2 to 1.25. In the poorest forms of compressor the value $n = 1.4$ is closely approached. For large, well-designed stage compressors and efficient intercooling, n, referred to the combined indicator cards (Fig. 39), may be as small as 1 15.

Work of Two-stage Compressors, without Clearance. The air is brought up to a certain pressure in one cylinder; passes to an intercooler, in which the temperature of the air is reduced, and finally enters a second cylinder, where the compression is carried to the desired terminal pressure. The cylinder ratio is such that the work is equally divided between the cylinders, but changes of conditions of operation other than those contemplated may destroy this equality. In the following discussion* it is assumed that the same quantity of work is done in both stages.

An inspection of the diagram, Fig. 37, shows that there must be some best intermediate receiver pressure, for which the total work of compression will be a minimum. For, if this receiver pressure approach either P or P″ (corresponding respectively to the points B and G on the compression curve), then would the compression approach single-stage work and the entire compression line would lie along BCG. But, with an intercooling receiver at any intermediate point, the broken line BCDE is followed, the saving in work over single-stage compression being represented by the area CDEG.

The net work of the compressor, W, represented by the area ABCDEF, is equal to the work of area ABCH of the first stage plus the work of area HDEF of the second stage, or $W = W_1 + W_2$. Let the condition of perfect intercooling be assumed;

* I desire here to acknowledge the valuable assistance of Dr. Charles E. Lucke, Professor of Mechanical Engineering in Columbia University, who kindly gave me the use of his original notes on the method of analysis employed in this discussion of the theory of stage compression and the effect of clearance in the different work cycles of adiabatic compression.

that is, the hot air discharged from the first cylinder is cooled in the intermediate receiver to the initial temperature of the intake air. The work cycle in each cylinder is the same as that of single-stage adiabatic compression, as expressed by equation (16), but with two additional symbols for pressures and volumes.

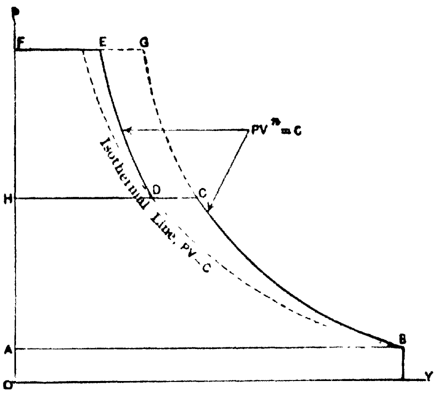


FIG. 37.—Reference Diagram, Two-Stage Compressor, with no Clearance and Perfect Intercooling

Let AB = V = initial volume of free air in first cylinder;
HD = V″ = initial volume of air in second cylinder;
OA = P = initial absolute pressure (atmospheric pressure);
OH = P′ = terminal absolute pressure in first cylinder, assumed to be also the intermediate receiver pressure and therefore the initial pressure in the second cylinder;
OF = P″ = terminal absolute pressure in second cylinder.

Hence:
$$\left.\begin{aligned} W_1 &= \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad . \; . \; . \text{ first stage} \\ W_2 &= \frac{P'V''n}{n-1}\left[\left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 1\right] \; . \; . \text{ second stage} \end{aligned}\right\} \; . \; . \quad (17)$$

But, assuming the intercooling to be perfect, PV = P′V″, whence:

$$W = \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} + \left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 2\right] \; . \; . \; . \; . \; . \; . \quad (18)$$

Since the best receiver pressure, P′, is that for which W is a minimum, by differentiating and placing the first differential coefficient $\frac{dW}{dP'}=0$:

$$\frac{dW}{dP'}=PV\frac{n}{n-1}\left\{\frac{n-1}{n}\,\frac{(P')^{\frac{n-1}{n}-1}}{P^{\frac{n-1}{n}}}-\frac{n-1}{n}\,\frac{(P'')^{\frac{n-1}{n}}}{(P')^{\frac{n-1}{n}+1}}\right\}=0$$

Whence: $\dfrac{(P')^{-\frac{1}{n}}}{P^{\frac{n-1}{n}}}=\dfrac{(P'')^{\frac{n-1}{n}}}{(P')^{\frac{2n-1}{n}}}$

or $(P')^{-\frac{1}{n}+\frac{2n-1}{n}}=(PP'')^{\frac{n-1}{n}}$, from which $P'=\sqrt{PP''}$, an expression for the best receiver pressure

Dividing both terms by P:

$$\frac{P'}{P}=\frac{(P\times P'')^{\frac{1}{2}}}{P}=\left(\frac{P''}{P}\right)^{\frac{1}{2}}$$

But, $\left(\frac{P''}{P}\right)^{\frac{1}{2}}=\frac{P''}{\sqrt{PP''}}=\frac{P''}{P'}$. Substituting these values in equation (18), remembering that $P'V''=PV$ and expressing the work in ft.-lbs.:

$$W=\frac{2\times144PVn}{n-1}\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}}-1\right]\quad . \quad . \quad . \quad . \quad (19)$$

which is the equation for two-stage compressor work, in terms of the initial volume and initial and terminal pressures, with perfect intercooling and best receiver pressure.

By a method similar to the above, the expression for the work of three-stage compression may also be deduced:

$$W=\frac{3\times144PVn}{n-1}\left[\left(\frac{P'''}{P}\right)^{\frac{n-1}{3n}}-1\right]\quad . \quad . \quad . \quad . \quad (20)$$

in which P‴ is the terminal pressure in the last, or high-pressure, cylinder.

Effect of Clearance in the Compressing Cylinder. In the preceding pages expressions are deduced for the work of compression with no allowance for the clearance volume of the cylinder.

From a mechanical engineering and structural point of view, the question of piston clearance is taken up in Chapter VII. It is necessary here to discuss the cycles of operation of single-stage and two-stage compression with clearance. While the work done per unit of air is the same as that shown by the general equations for isothermal and adiabatic compression, the work per unit of cylinder displacement will be changed, because of the re-expansion of the clearance air. In other words, clearance affects the volumetric output of the compressor, but not the work of compression per unit of volume of air taken into the cylinder.

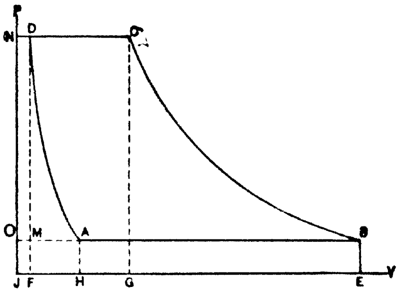


FIG. 38.—Reference Diagram. Compressor Working Isothermally, with Clearance.

Work of Compressors with Clearance. Isothermal Compression. Fig. 38 is a general reference diagram, in which BC represents an isothermal curve.

Let EB = MF = P;
GC = FD = P′;
JE = OB = V;
CN = V′;
JF = ND = clearance volume;
JH = OA = (OB − AB) = V‴, or volume occupied by the re-expanded clearance air.

According to the diagram areas:

Net work $\overline{ABCD}$ = compression work and delivery work $\overline{OBCN}$ − re-expansion work $\overline{OADN}$,

For the work of a compressor without clearance, and isothermal compression, the general expression, $W = PV \log_e\left(\frac{P'}{P}\right)$, has already been deduced. This applies to the areas bounded by the two horizontal lines, the vertical line and the compression line. Similarly, the re-expansion work represented by the area $\overline{OADN}$ (under the curve DA) $= PV''' \log_e\left(\frac{P'}{P}\right)$.

$$\text{Hence, } W = PV \log_e\left(\frac{P'}{P}\right) - PV''' \log_e\left(\frac{P'}{P}\right)$$

$$= P(V - V''') \log_e\left(\frac{P'}{P}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Replacing $(V - V''')$ by L, which represents the intake capacity of the compressing cylinder, neglecting heating during suction, and expressing P in lbs. per sq. in.:

$$W = 144PL \log_e\left(\frac{P'}{P}\right). \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Comparing equations (4) and (22) it is seen that they are identical, as noted above; but it must be remembered that the volume of air actually taken into the cylinder at each stroke and compressed is reduced on account of clearance, and hence the volumetric capacity of the compressor is also reduced. Moreover, neither in this work cycle, nor in that for adiabatic compression, is any account taken of the heating and cooling effects which occur during intake and compression, nor of frictional and other losses which affect capacity and work per unit of air. These points are discussed elsewhere in this chapter and in Chapters IV, V, VI, VII and X.

Single-stage Adiabatic Compression, with Clearance. The diagram, Fig. 38, may be used here also, by assuming the line BC to be an adiabatic curve. But, though the work areas are designated as above, under isothermal compression, and their significations are identical, their numerical values are different.

From Eq. (16), the work corresponding to area $\overline{OBCN} =$ $W_1 = \frac{144PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]$; and, similarly, the work corre-

sponding to the area $\overline{OADN}$ =

$$W_2 = \frac{144PV'''n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right].$$

Whence, $$W = \frac{144P(V-V''')n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad . \quad . \quad . \quad . \quad . \quad (23)$$

Replacing $(V-V''')$ by L (the intake capacity of the cylinder, with clearance):

$$W = \frac{144PLn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]. \quad . \quad . \quad . \quad . \quad (24)$$

Since V in equation (16) may be replaced by L, a comparison of this equation with (24) shows that the work is the same per unit of volume of air admitted to the cylinder; but, the volumetric output is reduced by the clearance.

Though the pressure-volume formulas serve for most purposes, it is sometimes convenient to have the work expressed in terms of cylinder displacement and volumetric efficiency:

Referring to Fig. 38:

Let D = displacement volume of cylinder in cu.-ft., or the effective area × stroke, represented on the diagram by MB = FE;

C = clearance expressed as a fraction of D; whence C × D = V_c = clearance volume, represented by ND = JF, and D (1 + C) = total cylinder volume in cu.ft., represented by JE;

V‴ = volume of re-expanded clearance air;

L = intake free air capacity = JE − JH = V − V‴;

E = volumetric efficiency $= \frac{L}{D}$ = the ratio of the length of the actual admission line, AB, to the total distance swept through by the piston.

Then: $V = D(1+C)$, and $V''' = V_c\left(\frac{P'}{P}\right)^{\frac{1}{n}} = CD\left(\frac{P'}{P}\right)^{\frac{1}{n}}$.

Whence $V - V''' = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right] = L.$

Substituting this value of $V - V'''$ in equation (23):

$$W = \frac{144Pn}{n-1} D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right], \quad . \quad (25)$$

which expresses the work in terms of displacement and clearance, with pressures in lbs. per sq. in.

Since $E = \frac{L}{D}$, $L = ED = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]$, as above; whence, substituting in (25):

$$W = \frac{144Pn}{n-1} ED\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]. \quad . \quad . \quad . \quad . \quad (26)$$

Work of Stage Compressors, with Clearance. (Fig. 39.) The method of analysis of the work of a two-stage compressor,

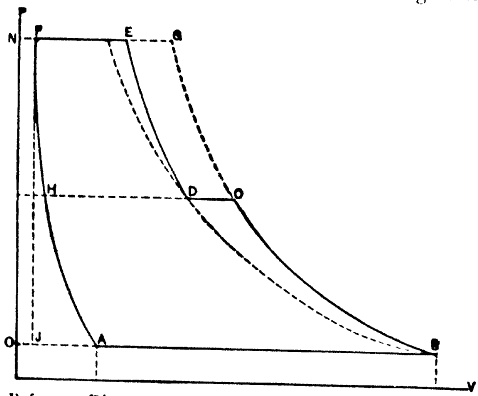


FIG. 39.—Reference Diagram. Work of Two-Stage Compressor, with Clearance.

without clearance (pp. 60–62), applies also here. With clearance the work for each stage is expressed by equation (23), the necessary changes being made in the signification of the different factors:

$$\left.\begin{aligned} W_1 \text{ (first stage)} &= \frac{144P(V - V''')n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \\ W_2 \text{ (second stage)} &= \frac{144P'(V'' - V^{\text{iv}})n}{n-1}\left[\left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 1\right] \end{aligned}\right\} \quad . \quad . \quad (27)$$

In equation (27) the symbols have the same significations as on pages 61 and 63, in addition to which V^{iv} is the volume of re-expanded clearance air of the high-pressure cylinder. But, if intercooling be perfect,

$$PV - PV''' = P'V'' - P'V^{iv},$$

that is, the weight of air entering the second cylinder is equal to that entering the first. Hence, the general equation for the work of a two-stage compressor, with clearance and perfect intercooling, takes the form:

$$W = \frac{144P(V - V''')n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} + \left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 2\right]. \quad . \quad (28)$$

By differentiating this equation, and making the first differential coefficient $\frac{dW}{dP'} = 0$, it is found that $P' = \sqrt{PP''}$. This is the value for the best (most economical) intermediate receiver pressure. It is the same as that deduced on page 62 for stage compression without clearance, since the receiver pressure is a function of the compression line and not of the re-expansion line.* By substituting in equation (27) the value of P' for best receiver pressure, it will be found that, as in the work cycle for stage compression without clearance, there is here also an equal division of work between the two cylinders. Finally, if the same value of P' be substituted in equation (28), this equation takes the form:

$$W = \frac{2 \times 144P(V - V''')n}{n-1}\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right]. \quad . \quad . \quad (29)$$

The diagram (Fig. 39) shows a single, continuous re-expansion line, FA, which is taken to represent the re-expansion lines of both cylinders. This evidently is true only when the clearances are proportionate; that is, when the *volume of the clearance air* of the high-pressure cylinder, after re-expansion, is equal to

* For the sake of brevity, the steps in the deduction of these and the following work formulas are omitted. Readers who desire to pursue the subject further will find a full discussion in Lucke's *Engineering Thermodynamics*.

the *clearance volume* of the intake cylinder. But the cylinders of stage compressors may, and usually do, have different clearances, between which no particular relation exists. However, since the volume of air delivered by the low-pressure cylinder must necessarily be equal to the volume received by the high-pressure cylinder, disproportionate clearance does not affect the work done per unit of air compressed. The diagram of the high-pressure is merely displaced somewhat with respect to that of the low-pressure cylinder, as shown by Fig. 40, in which FE = F′E′ and HD = H′D′.

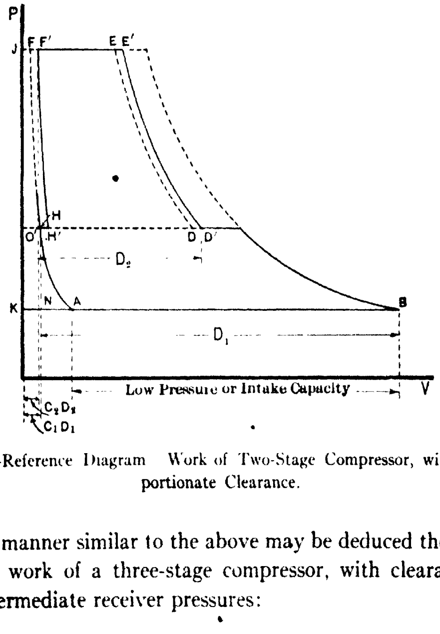


FIG. 40.—Reference Diagram Work of Two-Stage Compressor, with Disproportionate Clearance.

In a manner similar to the above may be deduced the expression for work of a three-stage compressor, with clearance and best intermediate receiver pressures:

$$W = \frac{3 \times 144 P(V - V''')n}{n-1}\left[\left(\frac{P'''}{P}\right)^{\frac{n-1}{3n}} - 1\right], \quad . \quad . \quad (30)$$

in which P‴ is the delivery pressure, and V‴ the re-expansion volume of the intake cylinder.

The work and capacity of stage compressors may also be expressed in terms of displacement and clearance.

Let D_1 and D_2 = cylinder displacements, respectively of the low- and high-pressure cylinders;

C_1 and C_2 = fractional clearances; whence $C_1 \times D_1$ and $C_2 \times D_2$ = clearance volumes, and $D_1(1+C_1)$ and $D_2(1+C_2)$ = total cylinder volumes, all in cu.ft.

$$E_1 = \frac{AB}{NB} = \text{volumetric efficiency of low-pressure cylinder;}$$

$$E_2 = \frac{H'D'}{O'D'} = \text{volumetric efficiency of high-pressure cylinder.}$$

In the demonstration leading to equation (25) it was found that $V - V''' = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]$. Applying the proper subscripts for two-stage work, and substituting this value of $V - V'''$ in equation (28), remembering that the volume discharged by the low-pressure is equal to that received by the high-pressure cylinder:

$$W = \frac{144Pn}{n-1} D_1\left[1 + C - C_1\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} + \left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 2\right]. \quad (31)$$

which expresses the work in terms of displacement, clearance, and initial and terminal pressures.

Since, when the intercooling is perfect, the temperature of the air entering the high-pressure cylinder is the same as that entering the intake cylinder, it follows that the product of the initial pressure in each cylinder and the volume of air admitted to each is the same for both cylinders. As previously stated, $P(V - V''') = P'(V'' - V^{IV})$; and, as shown on p. 62, $P' = (PP'')^{\frac{1}{2}}$ for best receiver pressure. Therefore,

$$P(V - V''') = (PP'')^{\frac{1}{2}}(V'' - V^{IV}); \text{ whence,}$$

$$\frac{V - V'''}{V'' - V^{IV}} = \frac{(PP'')^{\frac{1}{2}}}{P} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}.$$

But, since $V - V''' = D_1E_1$ and $V'' - V^{IV} = D_2E_2$,

$$\left(\frac{P''}{P}\right)^{\frac{1}{2}} = \frac{D_1E_1}{D_2E_2}. \quad \ldots \ldots \quad (32)$$

According to the reasoning which led to equation (26),

$$D_1E_1 = D_1\left[1 + C_1 - C_1\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right] \quad \ldots \quad (33)$$

and

$$D_2E_2 = D_2\left[1 + C_2 - C_2\left(\frac{P''}{P'}\right)^{\frac{1}{n}}\right]. \quad \ldots \quad (34)$$

Dividing (33) by (34) and combining with (32):

$$\left(\frac{P''}{P}\right)^{\frac{1}{2}} = \frac{D_1E_1}{D_2E_2} = \frac{D_1}{D_2}\frac{\left[1 + C_1 - C_1\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]}{\left[1 + C_2 - C_2\left(\frac{P''}{P'}\right)^{\frac{1}{n}}\right]}. \quad \ldots \quad (35)$$

For convenience in applying this equation, it may take the form:

$$\frac{D_1}{D_2} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}\frac{\left[1 + C_2 - C_2\left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right]}{\left[1 + C_1 - C_1\left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right]}. \quad \ldots \quad (36)$$

This transformation follows from the relation, for best receiver pressure:

$$\left(\frac{P'}{P}\right)^{\frac{1}{n}} = \left(\frac{P''}{P'}\right)^{\frac{1}{n}} = \left(\frac{P''}{P}\right)^{\frac{1}{2n}}.$$

Equation (36) expresses the ratio between the displacements of the cylinders, in terms of initial and terminal pressures and clearances.

If the percentage clearances of the cylinders are equal, $C_1 = C_2$ and $E_1 = E_2$; whence, the quantities in brackets of the second member of equation (36) cancel, and $\frac{D_1}{D_2} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}$. The same would be true if C_1 and $C_2 = 0$; a condition which may be assumed for compressors having very small clearance.

From equations (35) and (36), for the condition of best receiver pressure, several relations may be determined: (1) The ratio of compression for a given ratio of cylinder capacities, or conversely; (2) The ratio of cylinder displacements for known volumetric efficiencies; (3) The ratios of compression in the two cylinders which will produce best receiver pressure, the displacements and clearances being known, or conversely. In the third case, several approximations will usually be required.

For the performance of air compressors see Chap. X. Tables are there given, showing the work actually required per cu.ft. of free air, for single-, two- and three-stage compression.

CHAPTER IV

WET COMPRESSORS

ALTHOUGH wet compressors are obsolete in the United States, some attention should be given to them, because a few are still used in Europe, and a discussion of their design and operation will lead to a better understanding of the comparative merits of the modes of cooling employed in dry compressors.

Wet compressors comprise: 1. Those in which water is introduced in bulk into the air cylinder, and is injected also in the form of spray. 2. Those in which water is injected only in the form of spray or jets.

Compressors of the first type comprise some of the earliest forms. One of the best is the modernized Dubois-François, built at Seraing, Belgium. It has been widely used in Europe, for mining and tunnelling. Another is the Humboldt (Fig. 41). A mass of water forming the compressing piston is moved to and fro by a plunger *C*. Connected to each end of the cylinder by an easy curve is an air chamber, having inlet and discharge valves at *f* and *g*, made of rubber rings of round cross-section. As the piston reciprocates, the air is drawn alternately into one air chamber and compressed in the other. At the end of each stroke the air compressed by the rising mass of water in the chamber is discharged into the receiver. The air is partially cooled by contact with the water, and to keep the water cool proper circulation must be maintained. Further cooling is caused by the injection of sprays into the air chambers from a small force pump *c*, operated from the cross-head *d*.

Because of the inertia of the mass of water this type of compressor is generally limited to low piston speeds (100–150 ft. per min. or less in some cases). As this is about one-third to two-fifths of the piston speed of modern dry compressors,

a wet compressor is heavy and bulky for a given output of air. A more recent form of the Humboldt compressor is said to run successfully at speeds of 300–360 ft. per min., the temperature of the air at discharge being 77°–80° F. These remarkable results are open to question for regular, normal service. Lower speeds are always advisable for this type of compressor, as violent shocks are caused by running at high speed.

The Hanarte compressor is similar in principle to the Humboldt. Many have been built for French and Belgian mines, and also for ice-making plants. They are generally of large size, and are efficient at piston speeds of 250–275 ft. per min. The widely splayed out vertical ends of the cylinder cause

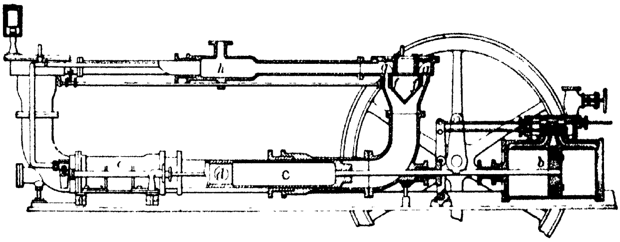


FIG. 41 - Humboldt Wet Compressor.

the water level to rise slowly towards the end of the stroke, and afford space in the cylinder heads for large and readily accessible inlet and delivery valves. Sprays are also used.

In wet compressors of this class an efficient circulation of water is difficult to maintain. Only a small quantity of cool water can be injected at each stroke, and without copious sprays the cooling is imperfect; although the mass of water in the cylinder and air chambers is large, there is between it and the air only a momentary surface contact. Since water is a poor conductor of heat, the air is cooled more by contact with the wet cylinder walls than with the small superficial area of the rising and falling water. Furthermore, the compressed air is practically saturated with moisture.

Compressors of the second type, in which cooling water is used only in jets or spray, are much less cumbrous than the older

design and permit a higher piston speed. They were first buil by Colladon, at the St. Gothard tunnel. Though some of then are still used in Europe, they are obsolete in American practice The air cylinder does not differ materially from that of the dry compressor. A water pipe is tapped into each cylinder head and fine spray is injected in front of the piston during compression. Since the water is in a state of fine division a relatively large surface contact is presented, and the air is thoroughly saturated with moisture during compression. Zahner states that Colladon's St. Gothard compressors, "which were run at a piston speed of 345 ft. and compressed the air to an absolute tension of 8 atmospheres (103 lbs. gage pressure), gave an efficiency which never descended below 80%, while the temperature of the air never rose higher than 12°-15° C. (53°-59° F.)." The temperature of the injection water is not stated, but must have been very low to obtain these results.

A dry compressor may be converted into a wet compressor merely by providing the water jets. The injected water collects in the cylinder until the clearance space at the end of the stroke is filled. The surplus is forced out at each stroke with the compressed air through the discharge valves, and is drained away from the receiver. As the piston clearance in well-designed compressors is very small, little water remains in the cylinder to be churned back and forth by the piston. The injection water should be pure and as cold as possible. Gritty water injures the cylinder, piston and valves.

In proper injection apparatus: 1. The injection must commence at the beginning of the stroke and continue to the end, against the advancing piston. 2. There should be thorough diffusion of the spray throughout the cylinder. By mere surface contact water takes up but little heat. Even a single strong jet is quite effectual, because on striking the piston it is broken into spray. 3. The volume of injected water should increase with the air pressure produced, that is with the quantity of heat generated. With insufficient water much moisture is taken up by the warm air and carried into the receiver.

The quantities of water required for different pressures are shown in Table III.*

TABLE III

Pressures.		Heat units Generated by Compression in 1 Lb. of Free Air	Pounds of Water to be Injected at 68° F. to Keep Final Temperature at 104° F.	
Above Vacuum, Atmospheres	Gage Pressure, Lbs.		Per Lb. of Free Air	Per Cu. ft. of Free Air
2	14.7	58.310	0.734	0.056
3	29.4	92.300	1.164	0.089
4	44.1	116.627	1.460	0.112
5	58.8	135.388	1.701	0.130
6	73.5	151.700	1.801	0.144
7	88.2	163.735	2.063	0.158
8	102.9	174.937	2.204	0.168
9	117.6	184.865	2.329	0.178
10	132.3	193.701	2.440	0.186
12	161.7	209.000	2.631	0.201

* This table is taken in part from that given by Zahner, "Transmission of Power by Compressed Air," p. 110. English units being substituted for French.

CHAPTER V

DRY COMPRESSORS

In dry compression no water enters the air cylinder except that which is carried as moisture in the air itself. All the cooling, aside from radiation, is effected by water-jacketing the cylinders.

Water-jackets. Fig. 42 shows the longitudinal section of a Nordberg cylinder. (Figs. 4, 8, 24, 26, 27, and other cuts of longitudinal sections, illustrate different types of jacketed cylinders.) The annular space JJ is occupied by water, and nearly one-half the area of each cylinder head is also covered by water jackets KK. The remainder of the end areas is occupied by the suction and delivery valves. Circulation of water is effected by pipes connecting with the openings A and B, respectively for inlet and discharge. To assist circulation the jacket spaces are subdivided. Cold water enters at A, and, after passing through the annular and end jackets JJ, KK, is discharged at B. For maximum cooling effect, the jackets on the cylinder heads surround the valves and air passages as completely as possible. C is a drain pipe for blowing out sediment.

In some designs, the annular jacket is divided by vertical partitions, so that the cold water entering at the top passes first around about one-fifth of the length of the cylinder nearest each end; then around the middle portion, and is discharged at the top. This arrangement recognizes that at the end of the stroke, where the air pressure is highest, the greatest amount of heat is generated. In other designs but little of the cylinder-head area can be jacketed, because of the space occupied by the inlet and discharge valves. This would seem to be a defect because, on approaching the end of the stroke, the piston rapidly

covers the annular jacket, leaving only a small part of its area available for cooling the hot air during discharge. It is at this point of the stroke that large end jackets are most valuable. In still other compressors, the water passes first into the end jackets, and then successively through separate compartments of the annular jacket. The delivery valves of some compressors are placed radially, close to the cylinder ends, whereby a

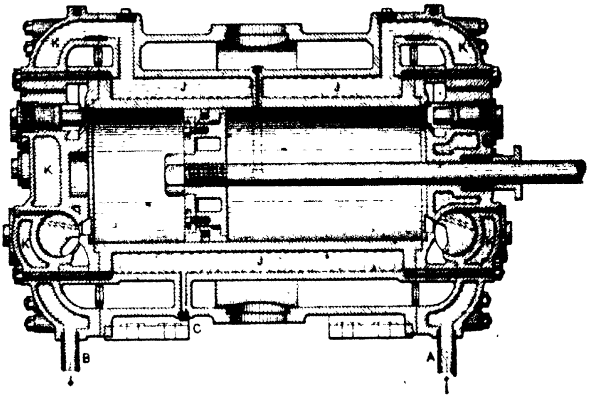


FIG. 42 Air Cylinder of Nordberg Compressor.

larger proportion of the heads can be jacketed (Fig. 43). The jacket in Fig. 44 has 8 longitudinal partitions, extending alternately from each end of the cylinder nearly to the opposite end. The water, which enters near the top, is forced to travel back and forth between the partitions and from end to end of the cylinder until it is finally discharged. The cooling water is often taken from a tank, set at an elevation above the compressor, or a small pump may be employed.

The useful effect of water-jackets depends largely on the running speed of the compressor. In the best single-stage compression, to say 70 or 75 lbs. and at not over 300 ft. piston speed, probably not more than about one-half of the total possible cooling can be effected; that is, n would be equal to, say, 1.25. Heat is generated faster than it can be abstracted,

since only a part of the air passing through the cylinder comes into direct contact with the cooling surfaces. The cylinder, discharge pipe, and even the receiver, are usually quite hot when the compressor is running at full speed; often too hot to be touched with the hand. With well-jacketed cylinders, and compressing only to 45 lbs., the temperature of the air at delivery has reached 280° F. The heat of compression in dry compressors probably ranges from 200°–400° F. for the

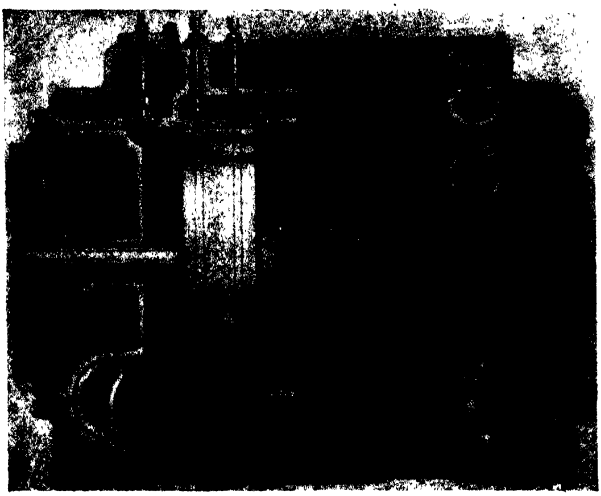


FIG. 43.—Air Cylinder of Allis-Chalmers Compressor, with Nine Delivery Valves Set Radially at Each End.

ordinary pressures used in mining, though not often exceeding 350°. The temperature should not be allowed to rise above this point.* At a mine in Montana, the writer has observed the thin wrought-iron delivery pipe of a 50-drill compressor red-hot for a distance of nearly 6 in. from the cylinder shell. Driving compressors at too high a speed often causes the poor results complained of by some users of compressed air.

* T. G. Lees, *Trans. Federated Inst. Mining Engrs.*, Vol. XIV, p 569. See also Chapter XIII of present volume.

The inner shell of the air cylinder, *i.e.*, between the cylinder and jacket, has sometimes been made of hard brass, which by its high conductivity assists in carrying off the heat. With the same end in view, the cylinder walls should be as thin as is consistent with safety. Besides cooling the air during compression, the water-jacket of a dry compressor is indispensable in keeping down the temperature of the cylinder shell. Without jackets the metal of the cylinder would become hot enough to burn the oil, and render proper lubrication impossible.

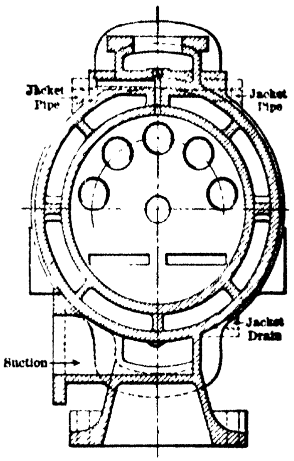


Fig. 44.—Air Cylinder, Class "E," Laidlaw-Dunn Gordon Compressor.

Piston Clearance in the Air Cylinder. Toward the end of the stroke, the compressed air in front of the piston begins to pass out through the delivery valves as soon as its tension exceeds that of the air in the discharge pipe to the receiver. But, in a dry compressor on completion of the stroke, a certain quantity of hot compressed air remains in the clearance space. On the back stroke this clearance air expands behind the piston, and no fresh air can enter through the inlet valves until the cylinder pressure falls below atmospheric pressure. Hence, it is never possible, in a dry compressor, to take a full cylinder of fresh air at each stroke; that is, the volumetric capacity per stroke, in terms of cu.ft. of free air, is always less than the volume swept through by the piston. In a wet compressor the clearance space is filled with water, and does not effect the volumetric capacity.

Fig. 45 shows the effect of clearance. Before the inlet valves can open, the piston must travel from c to b, and the cor-

responding cylinder volume passed through by the piston represents the loss of volumetric capacity. The actual effect of clearance on the volumetric efficiency of the compressor depends on the delivery pressure. The higher this pressure, the greater is the distance *cb*. The loss of volumetric capacity, although important in the operation of the compressor, does not involve a corresponding loss of useful work (see under Eq. 24, Chap. III). The compressed air remaining in the clearance space helps to overcome the inertia of the moving parts at the beginning of the return stroke, and to compress the air on the other side of the piston. The clearance air cools slightly during the momentary stoppage of the piston as the stroke is reversed, but the consequent reduction of pressure is negligible. In expanding behind the retreating piston, however, the clearance air cools rapidly and does not tend materially to raise the temperature of the incoming atmospheric air.

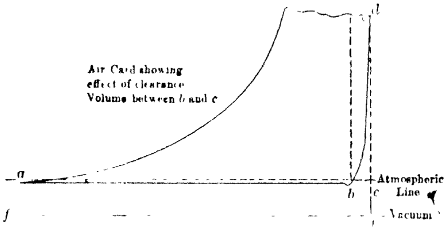


Fig 45

The effect of clearance in reducing the capacity of a dry compressor is shown by Fig. 46. For clearances greater than 1% the loss is serious, even at pressures of 75–100 lbs.

In cylinders of the same diameter and having the same amount of linear clearance, the ratio between cylinder volume and clearance volume depends on the length of stroke. This ratio is generally largest in short-stroke compressors and smallest in those of long-stroke. It varies, also, in compressors of different makers. Several Ingersoll-Rand compressors have the following ratios between cylinder and clearance volumes:

14-in. stroke	.0190	36-in. stroke	.0112
21 " "	.0176	48 " "	.0093
24 " "	.0126		

ranging thus from about 2% down to 1%. Some compressors of the same makers, of 42-in. stroke, but relatively small cylinder diameters, have clearances as small as .78, .80 and .90 of 1%, and several of 36-in. stroke have clearances of .83 and .84 of 1%.

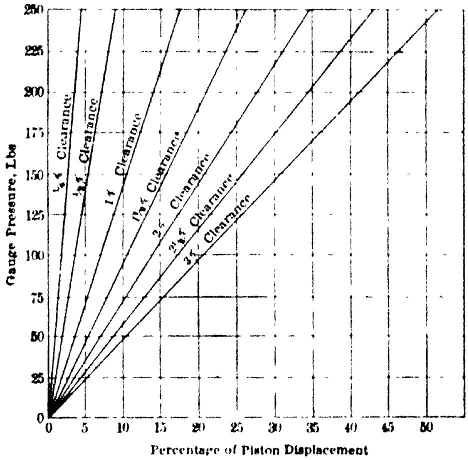


FIG. 46 – Diagram Showing Effect of Piston Clearance (*Eng. News*)

In some Laidlaw-Dunn-Gordon compressors clearances range from .75 to 1.25%. Clearances are generally larger; thus in the direct, electric-driven, two-stage, Ingersoll-Rand compressors, the following clearances are found in the intake cylinders:

28×24 in	2.15%	18×14 in	2.00%
23×20 "	1.70	17×14 "	2.19
19×16 "	1.85		

Compressors of some other makes have clearances as large as 2.5%. With few exceptions the lowest figures apply to large,

long-stroke compressors; the higher to the small, short-stroke machines.

Fig. 47 indicates the method of minimizing clearance for ordinary pistons, by casting a recess in the cylinder head to receive the piston-rod nut at end of stroke. With small clearances the compressor must have careful attention, so that, if the working length of the connecting rod should be varied in fitting new brasses, the piston will not strike the cylinder head.

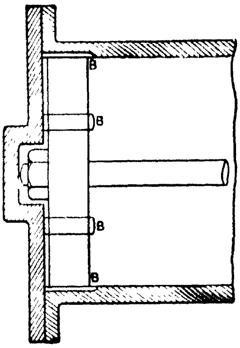


FIG. 47.

Examples of other devices for overcoming the disadvantages of piston clearance:

1. Longitudinal bye-pass grooves (Fig. 47, B) are cast in the inner surface of the cylinder near the ends, so that when the piston reaches the end of its stroke the grooves are partly uncovered, and the clearance air passes to the other side of the piston.

2. In slide-valve compressors the valve may be provided with a "trick-passage," which at the end of the stroke is brought into connection with two small ports entering the extreme ends of the cylinder, thus releasing the high-pressure clearance air into the other end of the cylinder.

An objection to releasing all the clearance air is that the sudden removal of the heavy pressure on the piston causes hurtful shocks. In recent American compressors the clearance space is very small, and the air confined in it is not released.

Dry *versus* Wet Compression. By bringing the air into direct contact with water the heat is most effectually absorbed, provided the injected water is properly applied (Chap. IV). Without cooling, the work converted into heat during compression, and therefore lost, is as follows:

Compression to	2 atmospheres,	9 2%	loss.
"	3 "	15 0%	"
"	4 "	19 6%	"
"	5 "	21 3%	"
"	6 "	24 0%	"
"	7 "	26 0%	"
"	8 "	27 4%	"

In well-designed dry compressors, working at 5 atmospheres, the heat loss is reduced about one-half (from 21.3% to 11%), while in ordinary mining practice, with single-stage compressors, the loss is often fully 15%. By spray injection this loss has been cut down to as little as 3.6%, and in some large, slow-running European wet compressors to 1.6%. But, low first cost and simplicity of construction may be more advantageous than a close approximation to isothermal compression. There are two considerations. (1) the effect of injected water upon the compressed air and the machines using it. (2) the effect of the water upon the working of the compressor.

First, by using large slow-speed engines, and an abundance of injection water, the air is well cooled, though at a higher first cost for plant. Wet compression gives a good indicator card. Table IV shows that in compressing moist air somewhat less work is expended than for dry air. This is because the specific heat of watery vapor is about twice that of dry air; therefore in the presence of moisture more heat is required to raise the temperature of the air in the compressing cylinder.

TABLE IV

Absolute Pressure, Atmospheres.	Gage Pressure Lbs	Ft.-Lbs. of Work to Compress 1 Lb. Air.	
		Dry Compression.	With Sufficient Moisture.
1	0		
2	14 7	23,500	22,500
3	29 4	37,000	35,000
4	44 1	48,500	45,000
5	58 8	58,500	52,500
6	73 5	67,000	60,000
7	88 2	75,000	66,000

Theoretically, a corresponding economy takes place when the air is expanded again in the machine using it.

Objections to Wet Compressors. The amount of heat absorbed during compression is proportional to the difference of temperature between the intake air and the injected water, and to the time of contact between the air and water. This difference of temperature is usually zero at the beginning of the stroke, reaching its maximum at the end. Hence: (1) to attain a fair approach to isothermal compression the piston speed must be very slow; (2) during the first part of the stroke but little heat is removed, and it is only when compression is complete, and discharge from the cylinder begins, that the cooling effect is at its maximum. At ordinary piston speeds, therefore, a large proportion of the total heat must be given up after the discharge valves have opened; in other words, after compression is completed. So far as economy of work is concerned, the lower final temperature due to spray injection is in a measure deceptive. The warmth of the air at discharge augments its moisture-carrying capacity, and the separation of the water in the receiver is of necessity imperfect in a receiver of any reasonable size. Much moisture passes into the air mains, deposits as the air cools in long pipe lines, and in cold weather may freeze so as to reduce the effective diameter of the pipe. Moisture remaining in the air has a further ill effect when it is used. At the instant of exhaust by the drill, or other air engine, the cold produced by expansion may cause troublesome accumulations of ice in the exhaust passages.

In the dry compressor, since air is a poor conductor of heat it can give up but little of its heat of compression between the piston strokes. But, although the moisture always present in atmospheric air will make its appearance as frost at the exhaust of the air machine, there is rarely enough of it to cause serious trouble.* The delivery of warm air by a dry compressor

* The quantity of moisture in the atmosphere, or its humidity, varies with the climate, the season of the year, and in a measure with the altitude above sea-level. It is usually greatest near the ocean or any large body of water. What is commonly called dry atmospheric air contains from 40 to 50% of the quantity necessary to saturate it. The degree of saturation in summer often reaches 90% or more.

is far less objectionable than warm air from a wet compressor.

Second, as to the effect of injected water upon the working of the compressor. Water in the air cylinder is always objectionable, because it makes lubrication difficult, causes rust, and increases the wear of piston and cylinder. There is no satisfactory method of lubricating wet compressor cylinders. Injected water must be pure and free from grit. Water that is harmless for use in jackets might be injurious to valves, cylinder and piston. Mr. W. L. Saunders states that, although the thermal loss is higher in dry than in wet compressors, the friction loss is considerably higher in the wet compressor. The net economy of the best wet compressors is probably no greater than that of the best dry compressors.

CHAPTER VI

COMPOUND OR STAGE COMPRESSORS

Compound or stage compressors divide the work of compression between two or more cylinders. In two-stage compressors air at atmospheric pressure is taken into the large or low-pressure cylinder, is there compressed to a certain point, and is then forced into the high-pressure cylinder, where it is brought up to the required tension. The cylinders are proportioned to divide the total work equally between them. This secures equalization of resistances, and promotes the efficiency of the cooling apparatus. The theory is given in Chap. III. Since the heat of compression increases with the pressure produced—though not proportionately—it becomes difficult at high pressures to keep down the temperature to a point permitting proper lubrication of the air cylinder. In attempting to compress even to 90 lbs. gage in one cylinder, the theoretical final temperature becomes 459° F. Though some heat is dissipated by radiation, the working temperature corresponding to this pressure may still be too high to be dealt with effectually by the ordinary water-jacket, because in a single cylinder the area to which cooling can be applied is too small relatively to the volume of air, and the total compression period too short. Even when working at moderate piston speed (350–400 ft. per min.), the cooling is so imperfect that the compressed air at delivery is very hot, causing considerable loss of pressure and work due to subsequent cooling.

Formerly, stage compression was employed for high pressures only, as for pneumatic locomotives, riveting machines, presses, compression of gases, etc. To produce very high pressures (500–1,000 lbs. or more) three- and four-stage compressors are necessary.

It is now recognized that two-stage compression is advantageous even for pressures of 70–80 lbs., as commonly used for machine drills. The cooling during compression is more thorough, because the total heat generated is divided between the cylinders; in each the temperature is lower than when the same total pressure is produced in a single cylinder, and the combined water-jackets afford a larger cooling surface.

A further cooling is effected by an "intercooler" (Figs. 49, 50), placed between the cylinders. It is an intermediate chamber, through which the air from the intake or low-pressure cylinder passes on its way to the high-pressure cylinder. The temperature of the air is here reduced, so that when the high-pressure piston begins its work the temperature of the volume of air on which it acts is considerably below that at which the air was discharged from the low-pressure cylinder. The total reduction of temperature depends on the volume of air under compression, the area of cooling surfaces and the length of time the air is in contact with these surfaces, which factor in turn depends on the piston speed.

Range of Work for which stage compression is applicable:

1. Although stage compression is theoretically advantageous for all pressures, it is of doubtful utility for gage pressures of much less than 75 lbs., because of the small saving as compared with the greater cost of the more complicated mechanism. It is generally applicable for pressures higher than 70–75 lbs.

2. Stage compression is especially useful for large plants, in which the percentage of saving will represent an amount sufficient to warrant the greater first cost.

3. The higher thermodynamic efficiency of stage compression is partly offset, and in poorly designed plants may be entirely neutralized, by the increased frictional losses due to the use of several cylinders. A well-designed, economical steam end should be used, together with efficient cooling for the air end; otherwise stage compression may cost more per cu.ft. of air delivered than simple compression in a well-designed compressor.

All ordinary stage compressors are double-acting; that is, on each forward and back stroke air is taken into the cylinders

on one side of the piston, while compression and delivery are going on on the other side. In the single-acting form the resistances in the cylinders are not as well equalized throughout the stroke. It is employed for some kinds of service, as for the high-pressure cylinders of locomotive chargers (Chap. XXVI), largely because of the difficulty of maintaining air valves for very high pressures; also to simplify the heavy castings required, and to give the water jackets more opportunity to act.

Double-Acting Two-Stage Compressors. This type is more satisfactory than the single-acting compressor, because the cycle of operations during each forward and back stroke is the same, and the distribution of resistances is more uniform. Three forms may be taken to represent accepted practice, *viz.*: the straight-line, two-stage compressor (Figs. 5–8) and the duplex forms, consisting of a pair of staged air cylinders, placed tandem to twin-simple, or cross-compound steam cylinders (Figs. 9–14). The last-named is best for large plants.

Fig. 48 shows diagrammatically a two-stage, straight-line compressor. Assuming that the pistons have reached the end of their forward stroke, the conditions in the two cylinders are approximately as follows: The intake cylinder (D) is full of air, practically at atmospheric pressure, while the high-pressure cylinder (G), with the intercooler (F) and connecting passages, are occupied by air just delivered from the intake cylinder, at, say, 30–35 lbs., or somewhat less than one-half the final pressure. On the reverse stroke the free air in front of the intake piston is compressed to 30 lbs. and delivered into the intercooler and high-pressure cylinder, while the air already occupying the latter is brought to the final pressure and discharged.

In standard tandem, two-stage compressors, the volumetric capacities of the low- and high-pressure cylinders are to each other in the ratio of about 10–4, the intention being to proportion them so that their ratios of compression are nearly equal. Thus the distribution of work and the heat generated in the cylinders will be equalized and most effectually dealt with by the intercooler. Practice as regards the relative volume of the intercooler and cylinders has changed greatly in recent years.

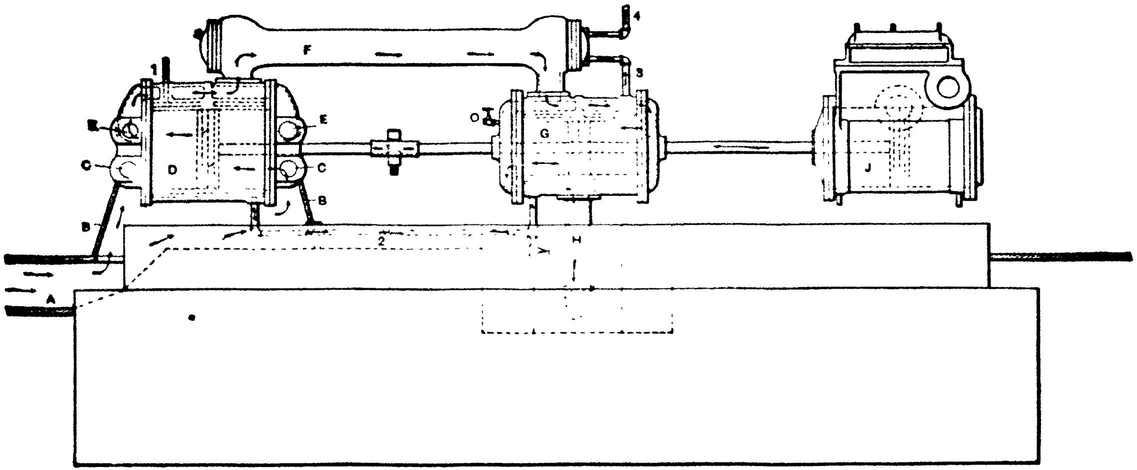


Fig. 48.—Diagram of Norwalk Two-Stage Compressor.

When pistons move as indicated by the arrow on the piston rod, the steam and air circulate in direction shown by arrows in the cylinders. Arrows on the water pipes show the direction of water circulation. A, inlet conduit for cold air; B, removable hoods of wood; C, inlet valve; D, intake cylinder; E, discharge valve; F, intercooler, G, high-pressure cylinder; H, discharge air pipe; J, steam cylinder; O, air relief valve, to effect easy starting after stopping with all pressure on pipes; 1, cold water pipe to cooling jacket; 2 and 3, water pipe; 4, water overflow or discharge. (Note.—This is not a recent design of the Norwalk compressor, but is used here as a diagram, as it well illustrates the working cycle of stage compression.)

In recognition of the importance of thorough intercooling, and the fact that the first cost of even a very large intercooler is moderate, while its maintenance is practically nil, it is now made of much greater capacity than formerly. The hot air from the intake cylinder is kept longer in contact with the cooling surfaces, because of its reduced speed of flow through the larger cross-sectional area of the intercooler, and it enters the high-pressure cylinder at a correspondingly lower temperature. The connections between cylinders and intercooler should be of as small volume as is consistent with reasonable frictional resistance to the flow of air through them; because the air occupying these passages at any given time is exposed to but little cooling save that due to radiation.

It may be assumed in good practice that, if the volume of the intake cylinder be 10, then the volume of its connection with the intercooler should be, say, 1.5, of the intercooler 4, of the connection to the high-pressure cylinder 1.5, and of the high-pressure cylinder 4 (the net capacity of the intercooler may be even greater than is here assumed). Having these proportionate volumes, the following cycle of operations will take place during a single stroke. Suppose this stroke to be from right to left, as indicated by the arrows in Fig. 48. By the previous stroke (left to right) the intercooler and its connections to the cylinders, representing a volume $= 1.5 + 4 + 1.5$, were filled with air compressed, at, say 30 lbs. This body of air was then shut off from both cylinders by their valves, and lost part of its heat and pressure by the action of the intercooler. During the first part of the following (left-hand) *stroke, the intake piston acts only on the cylinderful* of free air just taken in (volume = 10).* While this is being compressed, the advance of the high-pressure piston causes the air in the intercooler and its connections to begin to flow into the high-pressure cylinder, thereby increasing in volume and decreasing in pressure, until a point a little beyond mid-stroke

* The method of analysis here given is similar to that employed by Frank Richards, "Compressed Air," pp. 86–87, though the quantities used are taken to represent a closer approach to current practice in the proportions of the parts.

is reached. Beyond this point the pressure in front of the intake piston rises slightly higher than that in the intercooler and the delivery valves open, so that the piston acts upon the entire body of air: volume $=\frac{10}{2}+1.5+4+1.5+\frac{4}{2}=14$. Then, until the end of the stroke, both cylinders are in communication through the intercooler, *i.e.*, from the left-hand end of the intake to the right-hand end of the high-pressure cylinder, and an approximate equalization of pressure is established throughout.

Until the left-hand, intake delivery valves open, the air in the intercooler is isolated from the intake cylinder, in which compression has progressed without other cooling than that of the cylinder jacket. But when the warm air begins to pass through the intercooler into the high-pressure cylinder, the influence of the intercooler is exerted upon a new body of air. At the end of the left-hand stroke the closing of the delivery valves again shuts off the air in the intercooler from both cylinders. The high-pressure cylinder, on the right-hand side of the piston, is occupied by a body of air the temperature and pressure of which have been reduced by the combined effect of the intercooler and both water-jackets to a point below that due to the working pressure of the intake cylinder.

In the latter part of the left-hand stroke, when the intake delivery valves have opened and the piston of this cylinder is acting on the volume 14, as stated above, part of this air (volume $=\frac{2+1.5}{14}=25\%$ of the total) has passed beyond the influence of the intercooler, and another part (volume $=\frac{5+1.5}{14}=46\%$) has not yet reached it. At the end of the left-hand stroke the volume of air in the intake cylinder $=0$, in the intercooler and its connections $1.5+4+1.5=7$, and in the high-pressure cylinder 4, a total of 11, of which 1.5 has not reached the intercooler, but has been affected only by the water-jacket of the intake cylinder.

This analysis emphasizes the importance not only of employ-

ing a sufficiently large intercooler, but also of making the connecting passages small. The useful effect of the small intercoolers often used on straight-line compressors should not be exaggerated. The best economy is obtained only by cooling *during* compression and before the air leaves the cylinder. One-half of the total work of compression—that performed in the high-pressure cylinder—is done solely under such cooling influence as is afforded by the water-jackets of this cylinder. The jackets of both cylinders should, therefore, be of large area, with an efficient circulation of cold water. They should cover not merely the cylinder barrels, but as much of the heads as the space occupied by the valves permits. In the latter respect some compressor designs are deficient.

The details of the distribution of the air in the foregoing description apply exactly only to compressors in which the air cylinders are tandem to each other. In duplex stage-compressors, the cycle of operations is different because the pistons, instead of moving together in the same direction, work with one crank 90° in advance of the other.

Though the total work done should be equally divided between the air cylinders, still, by reason of the frequent variations in receiver pressure, upon which depends the actual terminal pressure of the high-pressure cylinder, an approximate equalization only is attainable. On the basis of some terminal pressure taken as normal, such diameters are assigned to the cylinders as will make their compression ratios equal, or nearly so. Take, for example, a pair of cylinders, 15 ins. and 24 ins. diameter, to produce 85 lbs. gage pressure. Assuming that the air between the stages is cooled to the original temperature, the absolute intake pressures of the cylinders will be inversely proportional to the squares of their diameters, or: $15^2 : 24^2 :: 14.7 : 37.64$. The absolute pressure of 37.64 lbs., as delivered by the intake cylinder, theoretically equals the intake pressure of the high-pressure cylinder. The ratio of compression in the intake cylinder is $\frac{14.7}{37.64} = 0.3905$; in the high-pressure cylinder, $\frac{37.64}{99.7} = 0.3775$. This is as close to perfect equalization as is necessary.

The Intercooler in its usual form is a long cylindrical chamber, containing parallel, thin brass or wrought-iron tubes, through which cold water is circulated. The air passes through the spaces between the tubes. The intercooler is placed in a convenient position between and usually above the cylinders (Figs. 8, 23), and as close to them as possible, so that the connecting passages may be short and of small volume, because, as already stated, the air in these passages at any given time is denied the cooling effect both of the cylinder jackets and of the intercooler itself. The intercooler tubes must be close enough together thoroughly to split up the body of air traversing the intermediate spaces and so secure the maximum cooling effect. It is intended that the temperature of the air, on passing from the intercooler to the high-pressure cylinder, shall be reduced nearly to the normal. The effect upon the compression curve of this drop in temperature is shown by Fig. 51; the high-pressure compression curve should, and often does, begin close to the isothermal line. Every 10% decrease in the temperature of the air delivered to the high-pressure cylinder decreases by about 1% the power required for compression. Thorough cooling has therefore been sought by increasing the volume and efficiency of the intercooler.

Brass intercooler tubes are preferable to iron because of their higher conductivity; but iron tubes cost less, and being rougher present a larger cooling surface to the air flowing between them. They should always be as thin as is consistent with the necessary strength. The tubes are expanded into tube-plates at each end, and by baffle-plates the air is compelled to pass through the entire volume of the intercooler. One tube-plate is attached to the shell, the other being free to move as the tubes expand or contract. The end water-heads are so divided that the water circulates actively back and forth several times, before final discharge. Fig. 49 shows a recent design. Air from the low-pressure cylinder entering at *A* passes alternately above and below the successive baffle-plates to the connection at *C* with the high-pressure cylinder. To compensate the decrease in air volume due to cooling, the baffles are spaced closer towards

the high-pressure cylinder; this maintains active circulation between the tubes and prevents excessive pressure drop. The

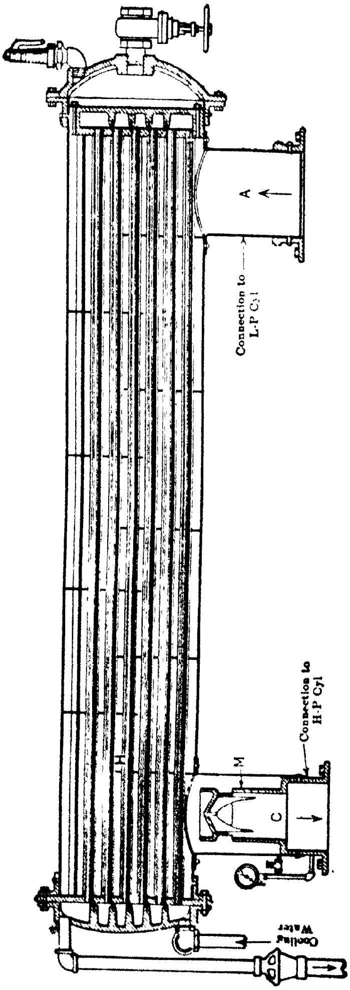


FIG. 49.—Ingersoll-Rand Intercooler, with Steel Shell and Cast-Iron Water Heads.

pressure drop allowed in this design corresponds to 3½ or 4 ins. of water column (=0.126–0.145 lbs. per sq. in.). The air may

be cooled to within, say, 18° F. of the cooling water temperature by using 2½ gals. of water per 100 cu ft. of air, or within 15° F. by 4 gals. per 100 cu.ft.* The baffle at *M* forms a trap for collecting the moisture deposited due to cooling, the water being drawn off by a drain cock.

The following comparison of the work done by single- and double-stage compressors shows the results of thorough cooling. Frictional losses are omitted in each case, and no account is taken of the cooling due to the cylinder water-jackets.

1. A single-stage compressor, producing a gage pressure of 70 lbs. at sea-level, with a 24-in. cylinder and a piston speed of 400 ft. per min., will have a capacity in terms of free air at normal temperature of 1,256 cu. ft. per min. For adiabatic compression, the mean cylinder pressure will be 33.83 lbs. and the H.P. 184.38.

2. For doing the same work in a two-stage compressor, having an intercooler capable of reducing the temperature of the air to the normal between the cylinders, it may be assumed that the intake cylinder has the same diameter, 24 ins., and that the pressure produced in it is 35 lbs. The mean pressure (adiabatic) corresponding to 35 lbs. terminal pressure is 25.6 lbs., and the H.P., 118.19. The diameter of the high-pressure cylinder, under the assumed conditions, is found by making the piston area inversely proportional to the increase in absolute pressure of the air delivered to it by the intake cylinder, *i.e.*, in the ratio of $14.7 : 35+14.7 = 1 : 3.38$. This gives an area of 135 sq. ins., or 13 ins. diameter. Compressing in this cylinder from 35 to 70 lbs. gage, the mean effective pressure will be 28.74 lbs., and the H.P., 46; or a total for both cylinders of $118.19+46 = 164.19$ H.P.

Compared with the power required for the same work in a single cylinder, this shows a saving of: $184.38-164.19 = 20.19$ H.P., or about 11%. The theoretically perfect cooling between the cylinders here assumed is unattainable, and the frictional loss in the stage compressor would probably be a little greater than in the single-cylinder machine; so that the net

* Charles A. Hirschberg, communication to the author.

gain due to intercooling may in this case be taken at, say, 7–8%. The saving is increased in dealing with higher pressures. (For "Stage Compression at High Altitudes," see Chap. XIII.)

Fig. 25, Chap. II, shows in some detail a Sullivan 3-pass intercooler. Fig. 50 shows an Ingersoll-Rand vertical inter-

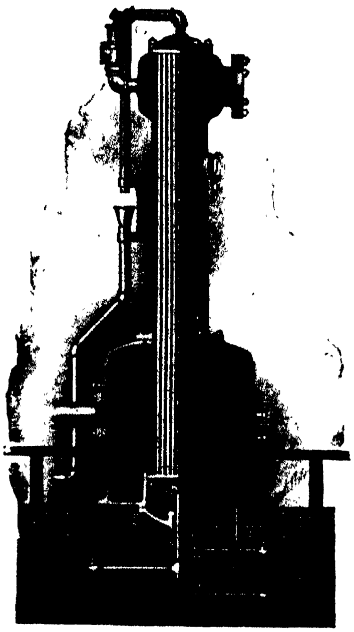


Fig. 50 - Vertical Intercooler. Ingersoll-Rand Co.

cooler, with all of its pipe connections. These coolers may be used also as "receiver-after-coolers," now considered as essential adjuncts of well-installed large plants (see end of Chap. XI) A similar appliance may be employed advantageously as an ante-cooler for the intake air.

Air Cards of Two-Stage Compressor. The compression curves of a two-stage compressor are shown in Fig. 51, the adiabatic and isothermal curves being also laid

down.* These cards (not accurately reproduced here) were taken from a pair of $7\frac{1}{2}$ and 14×16-in. cylinders, compressing to 110 lbs. gage, at 135 rev. per min., or 360 ft. piston speed. Initial temperature of cooling water, 55°; temperature at discharge from jackets and intercooler, 62° F. Several points are to be noted in connection with these cards:

First. The overlapping of the high- and low-pressure cards indicates a loss, because the work represented by the area of overlap is in reality *work done twice*. This is the result of the drop in pressure between the cylinders, caused by the resistance

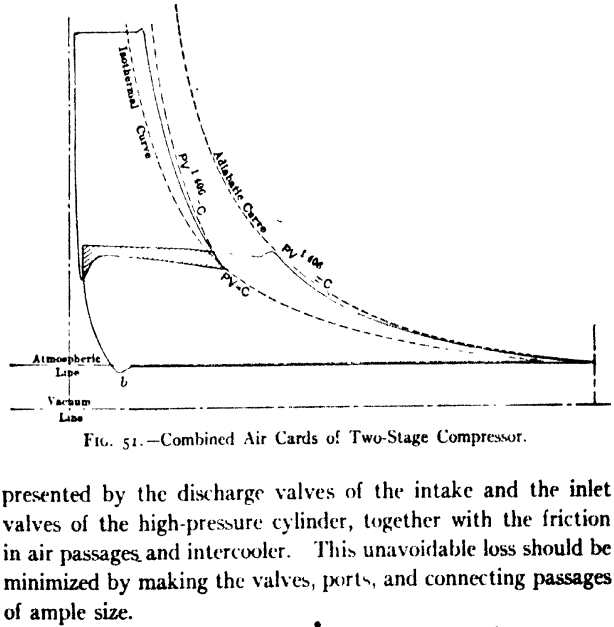


FIG. 51.—Combined Air Cards of Two-Stage Compressor.

presented by the discharge valves of the intake and the inlet valves of the high-pressure cylinder, together with the friction in air passages and intercooler. This unavoidable loss should be minimized by making the valves, ports, and connecting passages of ample size.

Second. As with single-cylinder compressors, the compression line of each cylinder of most stage compressors departs but

* This combined indicator card, which does not show all the minor irregularities in the lines, is from a Rand cross-compound compressor. It accompanies an article by F. A. Halsey, on "The Analysis of Air Compressor Indicator Diagrams," *American Machinist*, March 3, 1898, p. 158, and is reproduced here by permission.

little from the adiabatic curve. Aside from the thermodynamic advantage of dividing the total compression between two or more cylinders, and thereby lowering the average and final temperatures, it is the intercooler that must be relied on for the chief element in economical working. By its abstraction of heat the volume of air entering the second cylinder is reduced, so that $PV^{n=1.4}$ = constant becomes approximately $PV = C$, on beginning the second stage. But the compression line again rises rapidly from this point and continues not far below the adiabatic. Indicator cards from dry compressors which do not show approximately this relation between the lines are always open to suspicion. A leaky piston, for example, will lower the compression curve and make it appear that better work is being done than is really the case.

In constructing and reading a combined indicator card from a stage compressor, the adiabatic line applying to the compression in the second cylinder should be represented in its proper place (see Fig. 51). The complete graphic relation between the several heat curves is thus set forth.

Third. It is an advantage of stage compression that there is practically but one clearance space—that in the intake cylinder, and, as the air in this cylinder is at a low pressure, the resulting reduction in volumetric capacity is moderate, for the loss due to clearance is proportionately less for low than for high pressures. The piston clearance of the high-pressure cylinder cannot affect the volume of air delivered, because all the air discharged from the intake cylinder goes to the high-pressure cylinder and, barring leakage, must pass through it.

The heating of the cylinder walls and pistons reduces somewhat the working volumetric capacity of an air compressor, because, as the entering air is warmed, a smaller weight of it is taken into the cylinder at each stroke. Although the degree of this heating cannot be formulated, it is obviously less in a two-stage than in a single-cylinder compressor; for, aside from the effect of the intercooler, the smaller quantity of heat generated in each cylinder is more efficiently dealt with by their respective water-jackets.

CHAPTER VII

AIR INLET VALVES *

THE design and working of the inlet or suction valves greatly influence the compressor's efficiency. That there are still wide differences of opinion as to the best design is evidenced by the variety of types in use, and the lack of clearly defined distinctions as to their applicability under given conditions. In the older wet compressors, as the Dubois-François, various patterns of clack-valve were employed. Some are still used in Europe, like the elaborate, cam-controlled clack-valves of a compressor built by Schneider & Co., Creusot, France. For years poppet valves of numerous types held the field in the United States. They are furnished with springs, and are now actuated solely by difference of air pressure; though there have been examples of mechanically controlled poppets, as in the old Rand mechanical valve-gear.

While poppets are favored for some kinds of service, several other forms of inlet valve have been introduced. Modifications of the Corliss steam valve, first used in the Norwalk compressor, have been adopted for compressors of many other makes, as the Sullivan, Nordberg, Ingersoll-Rand, Laidlaw-Dunn-Gordon, and Allis-Chalmers. Besides these are the so-called "thin-plate valves," discussed later in this chapter.

Chief Requisites of Inlet Valves: 1. They must have a sufficient area of opening to permit free entrance of the air. 2. They must open readily near the beginning of the stroke, with minimum resistance, remain open until the end of the stroke, and then close promptly.

* This chapter is devoted chiefly to poppet valves and others which operate by difference of air pressure. For discussion of mechanically controlled inlet valves, see Chap. IX.

The point of stroke at which the inlet valves open depends on the piston clearance and terminal air pressure. Valves operated mechanically are sometimes incorrectly designed or set, to open exactly at the beginning of the stroke or a fraction later; in which case the clearance air is first exhausted through the valves and then, as the piston advances, the outside air begins to enter. This being so, little or no clearance would be shown on the indicator card. Premature closing reduces the volume of intake air, and hence the volumetric capacity of the compressor. Its effect on the indicator card is to lower the compression line near the beginning of the stroke, so as to approach the isothermal curve and make it appear that the compressor is doing abnormally good work.

As pointed out in Chap. III, although piston clearance reduces the volumetric capacity of the cylinder, it does not cause a corresponding loss of work, and it is of some benefit in assisting to overcome the inertia of the reciprocating parts of the compressor. Part of the work expended in compressing the clearance air is thus recovered, whereas, when the clearance air is exhausted by a premature opening of the inlet valves, the work represented by it is lost. The proper adjustment of spring-controlled poppets is a question of the strength of the spring, and since the effect of clearance varies with the terminal air pressure, the valves must be regulated accordingly. Any exhaust through the inlet valves is readily detected by the noise. When they are properly set, the compressor works more smoothly and the power consumed is slightly reduced. On the other hand, if the valves open too late in the stroke—due, for example, to a temporary reduction in working pressure—a little more power is required, this condition being shown by the slight drop in the re-expansion line at the point *b* (Figs. 45 and 51).

Inlet Area. The total area of the inlet ports varies greatly in compressors of different makers. It is sometimes as small as 4 or 5% of the piston area, running to a probable maximum of 15%. As the proper area is a function of the piston speed, it may be made less for slow- than for high-speed compressors. However, in one of the Leyner 2-stage compressors, with a 22-in.

intake cylinder and running at the moderate piston speed of 390 ft., the intake port area is 14.2% of the piston area. To prevent excessive frictional resistance during the inflow of air, the inlet area, under average conditions and for ordinary forms of valve, should be not less than 8 or 10% of the piston area. But if poppet valves are unnecessarily large, their inertia becomes great; and if too numerous, there are more parts to care for, and valuable water-jacket area on the cylinder heads is sacrificed. In the two-stage, straight-line, "Hurricane-inlet" compressors, of the Ingersoll-Rand Co., type AA-2, for cylinders from 15-in. to 24-in. diameter, the inlet area of the intake cylinder averages 13.2% of the piston area. For the high-pressure cylinders of the same compressors, the poppet valves have an average inlet area of about 11%. The inlet area of the duplex, two-stage compressors of the same builders, type "O-2," averages 13.6% of the piston area for both low- and high-pressure cylinders. In one type of the two-stage compressors of the Laidlaw-Dunn-Gordon Co. the percentage is 12–14.

Poppet Inlet Valves. A common form is the mushroom valve, Figs. 52 and 53. While the total inlet area should be ample, there are two special requirements in the case of ordinary poppet valves: (1) the area of each valve must be moderate, or the valve will be too heavy, causing unnecessary injury to the seat, and by its inertia too much resistance to the control of the spring; (2) the lift must be small, to insure prompt opening and closure, and to reduce "chattering," as well as wear. For these reasons the total area required is furnished by a number of independent valves, generally from 4 to 8.

The valve is of steel or bronze, with an easily removable bronze seat, the contact surfaces being ground true and the seating preferably coned. The stem works in guides, forming part of the valve casing, which is screwed into the cylinder head so as to be readily removed for adjustment or regrinding. Brass springs are used to avoid the effects of corrosion, and must be easily compressible to allow the valve to open under a small difference of pressure. The springs must be of the best material and calibrated to present no more than the minimum

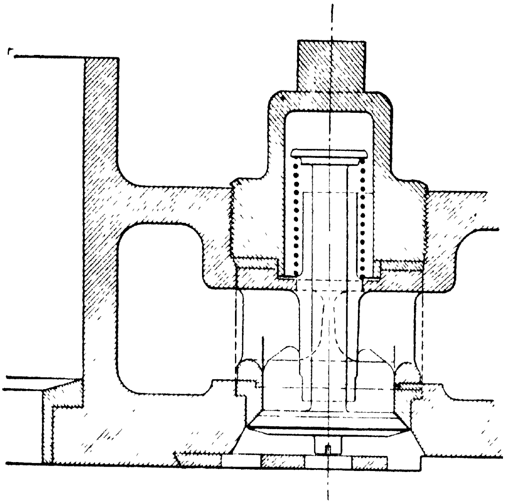


FIG. 52.—Norwalk Poppet Inlet Valve.

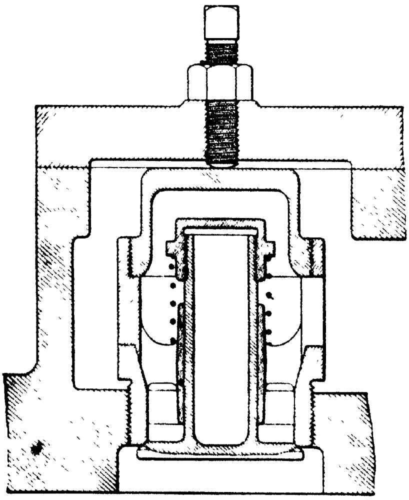


FIG. 53.—Laidlaw-Dunn-Gordon Poppet Inlet

requisite resistance, which varies from, say, 3 oz. to 8 or 10 oz. per sq. in. of valve area.

Ordinary poppets are opened by atmospheric pressure from without, when a certain degree of rarefaction of the air inside the cylinder has been produced by the movement of the piston; in other words, when the difference of pressure, after the clearance air has re-expanded, becomes sufficient to overcome the spring resistance. The loss of volumetric capacity due to spring resistance, in terms of free air, is rarely less than 2–3%, and is often more. At sea-level a spring pressure of 5 oz. per sq. in. of valve area causes a loss of about 2%. The diagram, Fig. 54, shows the effect of spring resistance in reducing the volumetric capacity of a compressor at different altitudes, from sea-level to 15,000 ft. elevation.

Spring-controlled poppets cause more or less irregularity in the entrance of the air, because, while the pressure of the outside air tries to open the valve, the spring tends to keep it closed. This often produces "chattering" or "dancing" of the valves, and has led to the introduction of various mechanical devices for definitely controlling them, as noted later. As the springs lose their original elasticity, and undergo alterations in strength, they require regulation from time to time; adjusting nuts on the valve stems are generally provided. If the springs be too slack, chattering increases; if too tight, the valves will open late in the stroke, and the air filling the cylinder may have a density less than that of the atmosphere. Aside from the spring resistance, the rate of inflow of the intake air varies with the speed of the piston. When its speed is greatest, at the middle of the stroke, the rate of inflow is at the maximum. While it is moving slowly, near the beginning and end of each stroke as the crank turns its centers, the relatively small negative pressure is insufficient to open the valves and keep them open against the springs.

The total valve resistance, including that due to throttling of the intake air and friction in passing through the ports, is kept as small as practicable, but can never be entirely eliminated. With some forms of inlet valves, other than spring poppets, the

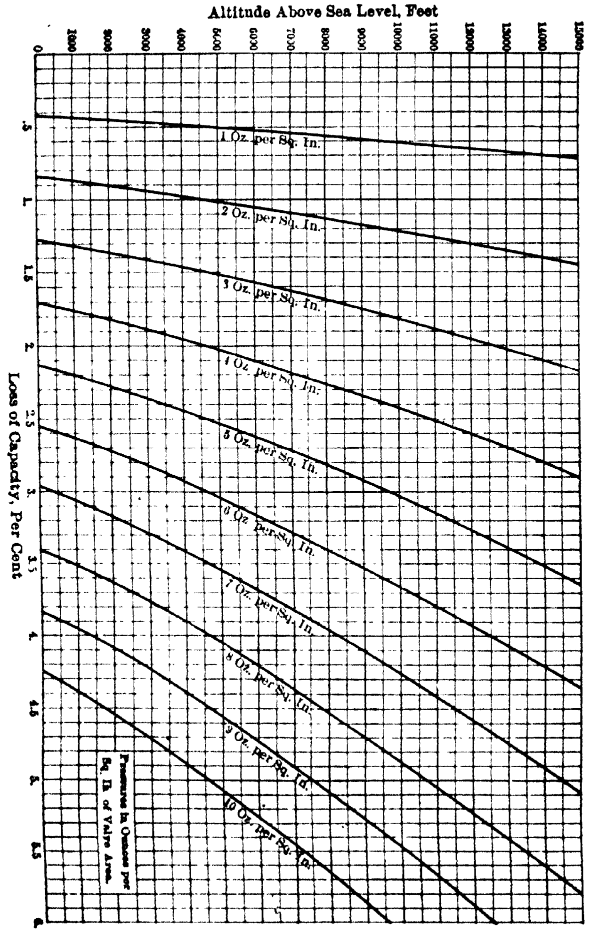


FIG. 54.—Effect of Valve Spring Resistance on Volumetric Capacity of Compressors. (*Eng. News.*)

resistance may be very small. Its usual effect is shown by the diagram, Fig. 55. There is generally sufficient resistance to keep the admission line, AC, at an appreciable distance below the atmospheric line, DE, throughout the stroke; the amount of loss from this cause is measured by the area of the indicator diagram lying below the atmospheric line. If the inlet area be too small or the valves poorly designed, the negative pressure may amount to 1 or 2 lbs. per sq. in. The point B, where the compression line crosses the atmospheric line, is the point of the stroke which must be reached by the piston before any useful work is done, and the volume passed through in travelling from

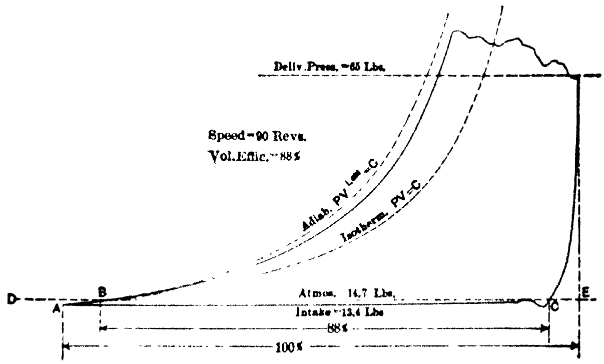


FIG. 55.

A to B represents the loss in volumetric capacity from this cause. The total loss of volumetric capacity, including that due to piston clearance, is represented by the length of AB+CE, and the volumetric efficiency of the compressor is equal to the length of the line BC, divided by the total length of the diagram.

Notwithstanding certain inherent disadvantages, the poppet valve is widely used, for both inlet and discharge. It is simple, easily regulated, and in case of leakage, due to cutting or unequal wear of the seating surfaces, is readily removed and re-ground. In stage compressors it is often used for the high-pressure cylinders, even when some other type is preferred for the low-

pressure. Poppets must be kept clean; they not infrequently cause trouble by sticking in their seats due to the accumulation of gummy oil, or they may be clogged by deposit of carbonaceous matter from decomposition of the lubricant, produced by excessive heating of the cylinder.

One form of Norwalk two-stage compressor has a special poppet inlet valve, for use when it is desired to employ air at two different pressures, obtained from a single compressor. In stage compression, though the air is actually produced at two pressures, of say 25-30 and 80-100 lbs. respectively in the low- and high-pressure cylinders, yet, if a part of the volume

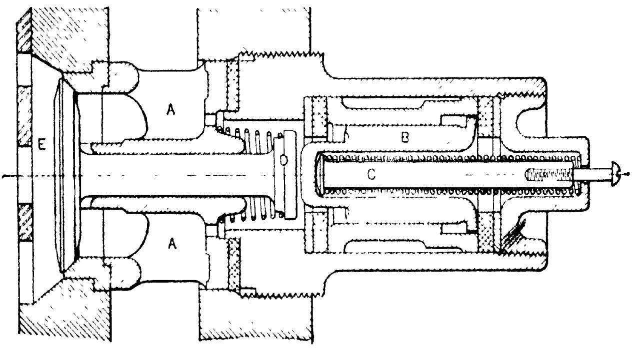


FIG. 56—"Skip-Valve." Norwalk Iron Works Co.

delivered by the intake cylinder be drawn from the intercooler, the high-pressure cylinder cannot work satisfactorily. The air remaining in the intercooler expands to a lower pressure before going to the high-pressure cylinder, so that the ratio of compression in this cylinder is increased, and the heat generated in it rises to a correspondingly higher degree. The rise in temperature produced by a considerable increase in the ratio of compression would prevent proper lubrication, and the conditions might be favorable for an explosion in the cylinder (Chap. XIV). This difficulty is met by using "skip-valves" (Fig. 56) as inlet valves of the high-pressure cylinder. They

open, and remain open, whenever the high-pressure inlet air falls below the normal, by reason of having drawn off a portion of the air from the intercooler. The high-pressure cylinder is thus partly unloaded, since the air entering at each stroke is returned to the intercooler. The skip-valve is a mushroom-spring-poppet DE, carried in the guides AA. Above the valve is a small spring-controlled plunger B, the space below which is occupied by air at intercooler pressure. When this pressure falls below that for which spring C is set, the plunger advances and forces open the inlet valve, holding it open until the intercooler pressure rises sufficiently to cause the plunger to recede. The valve is then free to work in the usual manner. The action of the valve thus adjusts itself to the varying pressure of the intake air coming from the intercooler.

Corliss Air Valves. As these are mechanically controlled, details are given in Chap. IX.

Ingersoll-Rand "Hurricane-Inlet" Valve, a modification of the old "Piston-Inlet" valve, is shown by Figs. 57 and 58. It has in turn been largely displaced by "plate valves," noted below. The piston is hollow and has a hollow back piston-rod for admitting the air. There are two large, ring-shaped valves (one in each face of the piston), of T cross-section. They are held in place, without springs or other connection, by guide-plates bolted to the piston faces. Their play is limited by these guides (see Fig. 58), which contain a series of circular ports, furnishing additional area for the passage of air. The valves are readily taken out for regrinding by removing the guide plates. At the beginning of each stroke, as the piston reverses, the valves alternately open and close by their own inertia. The valve in that face of the piston which is toward the direction of movement is always closed, while the other is open for the passage of the air entering through the hollow rod into the cylinder behind the piston. Because of the large diameter of the valves their throw, or lift, is small—in ordinary compressors say $\frac{1}{8}$ in. The area of the inlet tube is usually 13–14% of the piston area. Though the actual port area of the valve is less than this (about 8%), the velocity of flow is moderate, because, instead of having

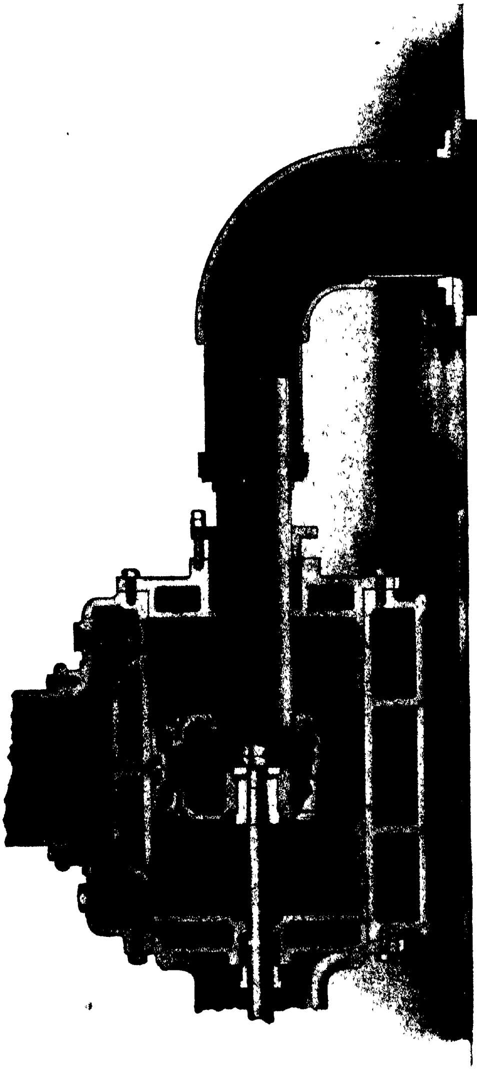


FIG. 57.—Ingersoll-Rand "Hurricane-Inlet" Valve. Section of Cylinder and Piston, Showing Valves in Position.

a group of inlet valves, the inlet is concentrated in a single annular opening.

The space in each cylinder head that would otherwise be occupied by inlet valves is utilized for additional water-jacket area. It has been objected that, since the inlet tube and piston are necessarily heated, the temperature of the intake air is raised in its passage into the cylinder; and that therefore the weight of air in the cylinder is relatively less than if it had entered by a more direct path. But, as the air enters in a

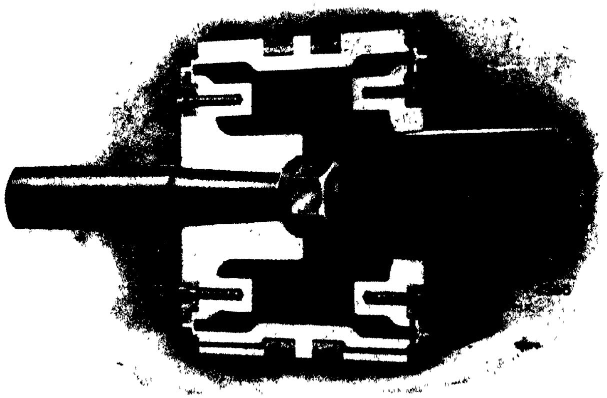


FIG. 58.—Ingersoll Rand "Hurricane-Inlet," Enlarged Section.

single large stream, instead of being divided into comparatively small areas of flow, it can absorb little heat until it reaches the valve, because only a thin film of the rapidly moving air in the inlet tube is in contact with the hot metal. Some heat is absorbed by the air in passing in the thin sheet through the valve port in the piston face; but thermometric observations, taken inside the inlet tube and piston, at speeds of 40 and 120 revs. per min., indicate a rise of not over 5° F. at the lower speed and even less at the higher. It is unlikely that better results are obtainable from either poppet or Corliss valves.

Plate Valves. For many years, "thin-plate" valves of several forms have been occasionally used; for example, the

Leyner (U. S.), the Guttermuth, and the Riedler (Europe), all now nearly obsolete. Since about 1913, interest in this type of valve has revived, and it is now used by a number of compressor builders (Ingersoll-Rand, Chicago Pneumatic Tool Co., Laidlaw-Dunn-Gordon, Sullivan, Allis-Chalmers, Robey, and Walker), for inlet or discharge valves, or both.

Plate valves appear to be satisfactory, though some think they have not yet been thoroughly tested as to endurance at the higher compressor speeds permitted by their small inertia. They probably lower the cost of manufacture.

The Guttermuth valve is a rectangular plate of thin steel with a grid seat. One side of the plate is coiled in a spiral, around a stationary spindle, the inner edge of the spiral being inserted in a longitudinal groove in this spindle. By placing several valves side by side any desired area of opening can be furnished. To avoid the harmful effects of inertia, the valves are very thin, with sensitive springs, and by so arranging them that the current of air in passing through the valve into the cylinder undergoes but slight changes of direction, serious eddying of the air around the edges of the plate is prevented.

Leyner Annular Valve is shown by Figs. 59 and 60. Though no longer made, some of the compressors using it are still in service. Fig. 59 is a longitudinal section through the adjacent ends of the low- and high-pressure cylinders of a straight-line, two-stage compressor, indicating incidentally the circulation of the air through the intercooling tubes. The inlet and discharge valves differ only in size. The valve (Fig. 60) is a thin steel plate cut in a peculiar form. The outer, or seating portion, is a narrow annulus, with two arc-shaped strips terminating in a central ring, which is locked to the cylinder head by a nut encircling the piston-rod. The arc strips, connecting the seating part of the valve with the fixed center serve as springs. There is but one inlet and one delivery valve at each end of the cylinder. The inlet ports, DD (Fig. 59), 4 in number, are curved, slot-like openings. There are 6 similar but smaller discharge ports,

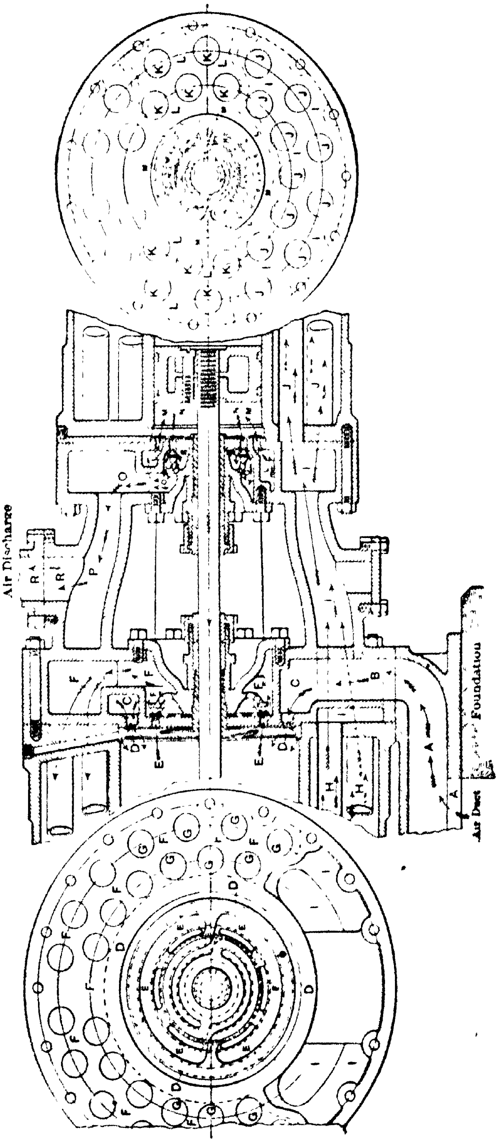


FIG. 59.—Leyner Compressor. Part Section, Showing Flat Annular Inlet and Delivery Valves.

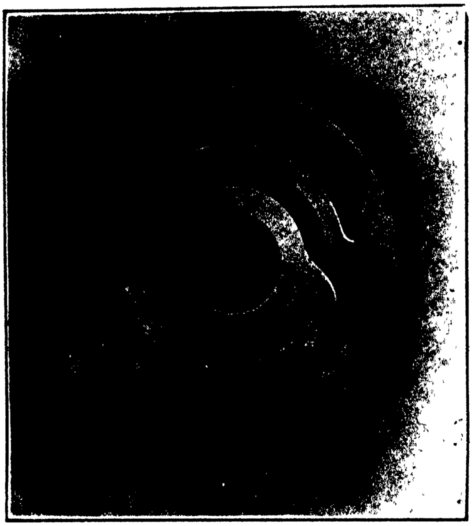


FIG. 60.—Leyner Annular Inlet Valve.

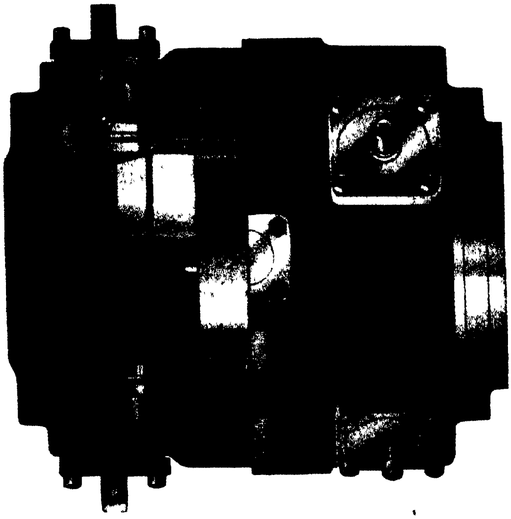


FIG. 61.—Laidlaw-Dunn-Gordon Air Cylinder, with Valves in Place

EE. Total area of inlet ports is about 14%, and of discharge ports, nearly 9% of the piston area.*

Laidlaw-Dunn-Gordon "Feather Valve." Fig. 61 shows the cylinder, with the inlet and discharge valves, which are

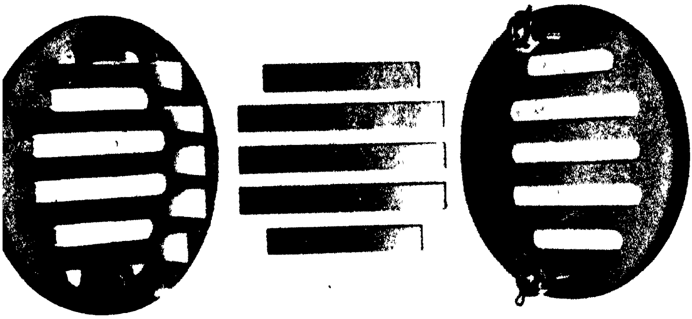


FIG. 62 Laidlaw-Dunn-Gordon "Feather" Valve, with Cover Removed.

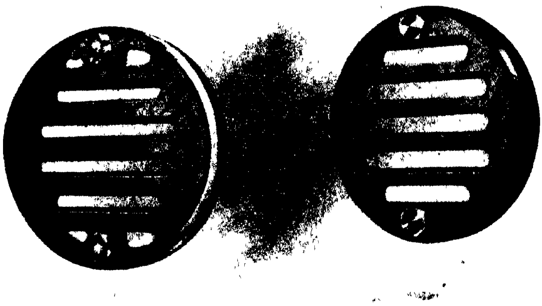


FIG. 63.—Laidlaw-Dunn-Gordon "Feather" Valve, Assembled.

alike and interchangeable; Fig. 62, the valve with cover removed; Fig. 63, the bottom and top views of the assembled valve. Figs. 1 and 24 (Chap. II) include sectional views of

* Tests have shown an intake pressure loss of 0.9 oz. On a card with a 20-scale spring, this would be represented by a difference of only 0.003 in. between the intake and atmospheric lines.

the valve, with its setting in the cylinder. The valve consists of 5 thin rectangular steel strips, the middle portions of which are free to lift from the seat to permit the passage of air. When the valve opens, the ends of each strip remain in contact with the seat, being retained in place between the valve cover and seat, though not held rigidly. As the steel strips are very light, the destructive effects of inertia are minimized. Fig. 24 shows the yoke and set-screw by which the valve cover is held in place.

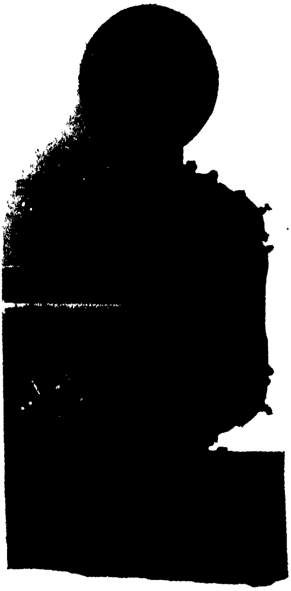


FIG 64.—Cylinder and Intercooler of Ingersoll-Rand Class "ORG" Compressor, with Plate Valves.

Ingersoll-Rand Plate Valve (Figs. 64, 65, 66). Referring to Figs. 65, 66, the valve F is a perforated disk of thin plate. The outer or seating portion is connected to the central ring N (firmly attached to seat A by stud bolt B) by the arms MM. Thus the valve can not turn on its center, but always rests in the same position. Arms M are ground thinner than the valve itself, to secure elasticity. Thickness of a medium size valve, 0.07 in.; diameter varies from 4.5 to 15 ins. The four light springs H of the cushion plate or buffer hold the valve on its seat A, when at rest, against the slight resistance of MM. When the required air pressure is reached, the valve rises against springs H to its full opening. Fig. 66 is a section of the valve assembled: The valve and cushion spring are separated by the washers G and E, and are attached to the seat by the bolt B. As there are no rubbing guides, lubrication is unnecessary. The port area is large;

height of lift is from 0.08 to 0.14 in., according to size of valve. Weight of the valve is about one-third that of an equivalent poppet valve. The inlet and discharge valves are identical.

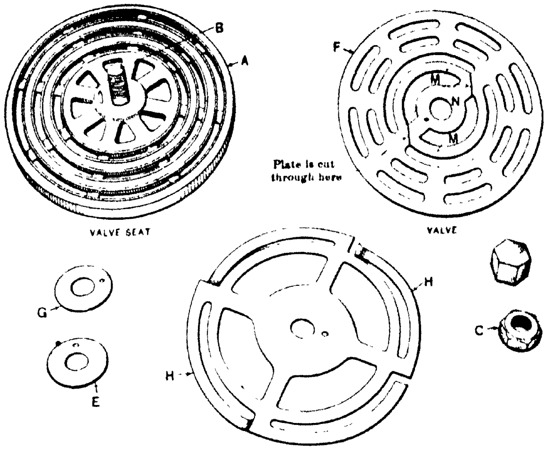


FIG. 65.—Parts of the Ingersoll Rand Plate Valve.

This valve is now being extensively used in the compressors of the Ingersoll-Rand Co. It somewhat resembles the older Leyner annular valve (Figs. 59 and 60).

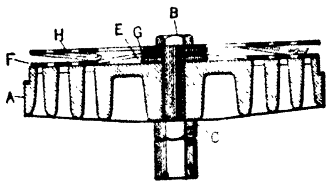


FIG. 66.—Section of Ingersoll-Rand Plate Valve, Assembled.

Sullivan Plate Valve ("end rolling finger-valve") was applied in 1917 to the angle-compound compressors (see Fig. 25, Chap. II) of these makers. Fig. 67 shows the design of valves and

seats in the low-pressure cylinder, for both inlet and discharge; Fig. 68, the position of the valves in the end of the cylinder. The valve consists of a group of thin, flat strips ("fingers"), of spring steel, with grid seats. Over the fingers is a corresponding row of curved guard plates. As the fingers open they roll against their guard plates from the stationary end towards the tip, and conversely on closing.

In the high-pressure cylinder, the inlet and discharge valves are also of the plate type, but are placed in cages set radially

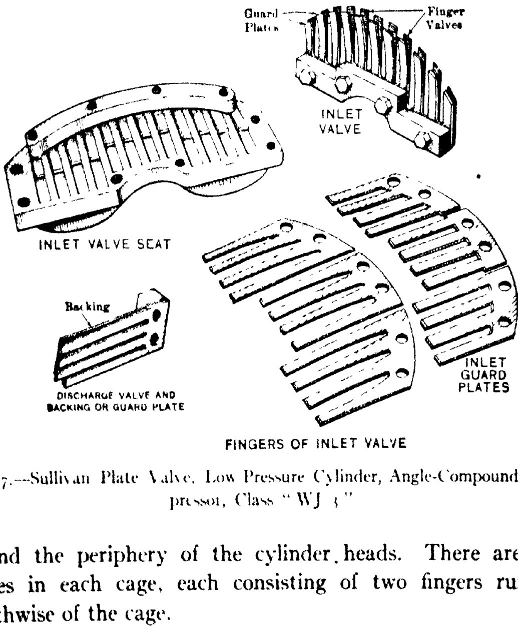


FIG. 67.—Sullivan Plate Valve, Low Pressure Cylinder, Angle-Compound Compressor, Class "WJ 3"

around the periphery of the cylinder heads. There are two valves in each cage, each consisting of two fingers running lengthwise of the cage.

Arrangements for Conveying Inlet Air to the Compressor. The colder the intake air the smaller the volume occupied by a given weight of air taken into the cylinder, and the greater the volumetric output. Mr. Frank Richards states:* "The volume of air at common temperatures varies directly as the absolute

* "Compressed Air," p. 55.

temperature. With the air supply at 60° its absolute temperature is 521°, and its volume will increase or decrease $\frac{1}{521}$ for each degree of rise or fall of temperature. Therefore, if in securing the supply of air we can get a difference in our favor of 5° . . . ., we accomplish a saving of about 1%. If a difference of temperature of 10° can be secured 2% is saved," practically without cost. The practice of taking air from the engine-room is a common one at mines, and is bad not only because such air is usually heated to a considerable degree, but is apt also to be charged with dust, which causes unnecessary wear of valves

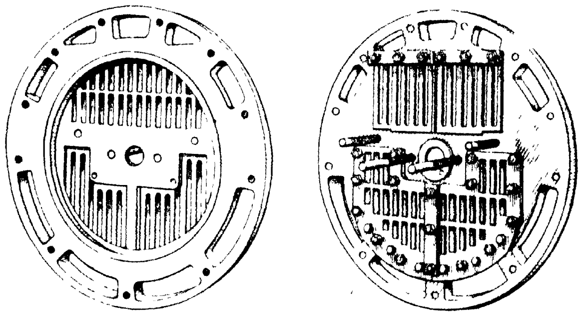


FIG. 68.—Sullivan Plate Valves in Place on Inner Head of Low-Pressure Cylinder, Angle-Compound Compressor, Class "WJ 3."

and piston. Fresh air should be taken, preferably from some point outside of the building. A box or pipe of wood is better than one of iron, because of the smaller conductivity of wood. Its cross-section should be sufficient, say, at least one-half the area of the cylinder, to avoid loss from friction. To make such a connection conveniently, the inlet valves should be enclosed in an external air chest on each end of the cylinder. Compressors having a single inlet valve are well adapted for making this arrangement. Care should be taken to prevent the entrance of dust, leaves, or rubbish. If the inlet is open, particles floating in the air may be drawn in and obstruct the valves or injure

their seats and the working surfaces of piston and cylinder. By building a conduit from the outside of the compressor house to the air box enclosing the inlet valves, a greater saving can be effected in winter than in summer, but even in warm weather some advantage is gained, especially if the conduit opens on the north side of the building, out of reach of the sun's direct rays, and is carried vertically to some height above the ground. In the Ingersoll-Rand "Hurricane inlet," the outer end of the hollow back piston rod is inclosed by an intake duct (Fig. 57).

CHAPTER VIII

DISCHARGE OR DELIVERY VALVES

THE conditions affecting the action of the discharge valves of a compressor are wholly different from those governing the inlet valves. While the latter must open under very small differences of pressure, the discharge valves are subjected to a heavy pressure on both sides. Furthermore, owing to irregularities in the use of the air, the receiver pressure usually fluctuates considerably, so that the point of the stroke at which the discharge valves open cannot depend solely on the conditions, as to the ratio of compression, etc., under which the compressor is working. The time of opening must depend also on the relation between the variable pressures in cylinder and receiver. Hence, the sensitiveness of operation essential in inlet valves is unnecessary for discharge valves. The chief requirements are that they shall be strong, free to open when the cylinder pressure exceeds that of the receiver, have strong springs so as to close promptly at the end of the stroke, fit accurately on their seats, have large free air passage around the rim of the valve, and ample guiding surface to insure accurate alinement in their movements. Delay in closing, or leakage between valve and seat, are far more serious than for inlet valves, because these defects are equivalent to an increase of piston clearance and consequent reduction in the volumetric capacity of the cylinder. The leakage of even a small quantity of compressed air back into the cylinder is equivalent to the loss caused by an abnormally large clearance space. Compared with inlet valves, therefore, discharge valves afford a relatively limited field for innovation or improvement.

Poppet Discharge Valves. Except the "plate valves" (Chap. VII), and a few designs in which mechanical control is introduced (Chap. IX), nearly all discharge valves are cup

shaped poppets, with internal springs. Though varying in details, they are represented fairly by Figs. 70 and 71. Other designs are shown in the sections of air cylinders illustrated in

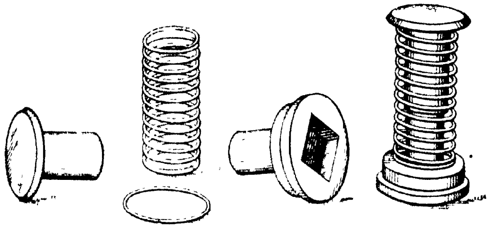


FIG. 69.—Ingersoll Rand "Imperial" Poppet Discharge Valve.

Chaps. II and VII. They are more rarely of the mushroom type, like that in Fig. 69, because these may not afford sufficient guiding surface. Valves are of steel or bronze, with bronze

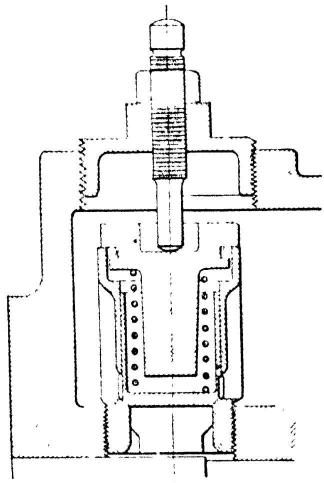


FIG. 70.—Laidlaw Dunn-Gordon Poppet Discharge Valve.

seat. To assist keeping them tight, the seating surfaces are coned. A group of poppets are commonly employed, to avoid making them of large size and weight. Under their high work-

ing pressure, the inertia of heavy valves causes destructive wear. Each valve should be readily accessible for adjustment, regrinding, or renewal. They are therefore covered by caps screwed into the cylinder head; or, in some makes, by cover plates on the valve chamber.

Cataract-Controlled Poppets. In these the valve not only has a spring, but its action is further modified by attaching the valve stem to the piston of a small cataract cylinder, containing air or oil, to ease its movement and avoid hurtful shocks.* These valves are employed to a considerable extent

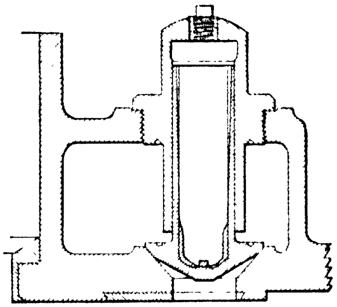


FIG. 71 —Norwalk Poppet Discharge Valve.

in Europe, but not in the United States. Some are satisfactory for slow piston speeds; for high-speed compressors they do not work with sufficient promptness to prevent " slip " or leakage of compressed air back into the cylinder. Their chief object may be attained in another way, as by the partial control of an accompanying Corliss valve (see Fig. 75, Chap. IX).

Riedler Discharge Valve (Fig. 72) represents one of several patterns employed in the Riedler compressors. It is a light, cylindrical valve A, with packing rings D. The cylinder in this case is vertical, and the piston L carries at its periphery plate P, held in place by stud N and spring M. When closed the valve seats on plate E, being held against it by the air

* Cataract-controlled poppets are employed by some European compressor-builders for both inlet and discharge.

pressure in the discharge passage acting on the *under* side of the upper flared end. In this position the round air ports near the lower edge of the valve are closed by the valve guide, at CC. As the piston advances, and when the cylinder pressure exceeds that in the receiver, the valve is opened by the air pressure on the *upper* side of the flared end. This movement of the valve is cushioned by the air trapped above the guide BB. On reaching the end of the stroke, the plate P, on the piston, strikes the lower edge of the valve and closes it against its seat E, the shock being cushioned by springs F and M. The double

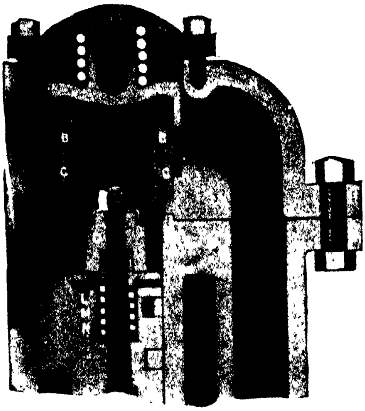


FIG. 72.—"Express" Poppet Valve, Riedler Compressor.

cushioning, in both opening and closing, tends to durability; and moreover, when plate P strikes the valve, the crank is nearly on its center, so that the piston is moving very slowly. Mechanically controlled air-valves of the Riedler design are described in Chap. IX.

"Thin-Plate" Discharge Valves are being used in some recent types of compressor. They are identical with the corresponding inlet valves. For examples, see Figs. 61–68, Chap. VII.

Several other forms of discharge valve are noted in Chap. IX, in connection with mechanically controlled valve motions.

Discharge Area for Air Cylinders. The volume of air to be discharged having been reduced by compression to a small

fraction of the volume at atmospheric pressure, it might appear that the total discharge area could be made much smaller than the inlet area, without causing excessive frictional resistance. But the compressed air must be forced out of the cylinder in a relatively short period of time. While the inlet valves are open throughout nearly the entire stroke, the delivery must take place while the piston is making, say, the last third of the stroke. Therefore, in a compressor of ordinary design, with several poppet inlet and discharge valves, the total discharge area should be about equal to the inlet area, provided the piston speed be moderate. When the inlet area is concentrated in a single valve (for example, the Ingersoll-Rand "Hurricane-Inlet"), the discharge area is about double the inlet area, though this relation varies in cylinders of different sizes, being proportionately greater in the larger compressors. Other things being equal, the discharge area should increase with the piston speed. For a speed of 350 ft. per min., the discharge area may be, say, 10% of the cylinder area; for speeds of 450-600 ft. per min., 15%. In some compressors, the discharge area is as small as 8.5-9.5%.

The above considerations apply also to the passages through which the air passes from the discharge valves to the pipe leading to the receiver. In some designs these are too restricted to permit a free flow of the air. The velocity of discharge should be kept low to minimize frictional resistance; otherwise, during the delivery period the pressure in the cylinder will rise momentarily above the normal, and then drop back after the air has passed out to the receiver. This causes loss of power and unnecessary strains on the moving parts of the compressor. The amount of this loss is represented by the irregular area of the air card which lies above a horizontal line drawn through the point corresponding to the pressure at the end of delivery (Fig. 55). When the discharge valves first open, the piston is moving at a high velocity, and equilibrium with the receiver pressure is only attained as this velocity decreases toward the end of the stroke.

CHAPTER IX

MECHANICALLY CONTROLLED VALVES AND VALVE MOTIONS

Mechanical Control for Inlet Valves. The disadvantageous features of inlet valves that depend primarily for opening and closing upon difference of air pressure have led to the introduction of numerous mechanically controlled valves. By their use fewer valves are required, because they may be made larger and have a higher lift. They are controlled by being in some way connected with the rotary or reciprocating parts of the compressor. By thus modifying the effect of the valve spring, a prompter opening is secured, so that the compressor takes more nearly a full cylinder of air at each stroke.

In some designs the connection between the valves and their operating mechanism is positive and fixed for any one setting of the valves, which are timed with respect to the piston stroke, to open at the instant the clearance air has been re-expanded to atmospheric pressure, and to close at the end of the stroke. In other designs springs are used in connection with the controlling mechanism, thus allowing for variations in working conditions, as well as for inaccuracies of adjustment or slight derangements caused by wear of parts. Still other valve motions exert a partial control, which, within narrow limits, leaves the valve free to act under difference of air pressure inside and outside of the cylinder.

As a rule, the inlet valves only are positively controlled, and in most cases the Corliss type of valve is used. While mechanically controlled valves may be employed for the intake cylinders of stage compressors, they are not suitable for the high-pressure cylinders; the inlet valves of these are subjected to heavy pressure on both sides, and are best allowed to open

and close solely under the difference between these pressures, which is more than sufficient to produce prompt action of the valve at the proper time. Poppet valves are therefore generally used for this service.

Mechanical Control for Discharge Valves is generally unsatisfactory, because of the fluctuations of receiver pressure. In attempting to open them by a positive mechanical movement, at a fixed point of the stroke, two cases may occur: (1) in event of a drop in receiver pressure below the normal, the valves and their controlling mechanism would be subjected to a heavy strain, before the point of opening is reached, due to excess of cylinder pressure; and, (2) if the receiver pressure should rise above normal, the valves would be held forcibly on their seats, by the excess of receiver pressure, after being released by the controlling gear. In either case, derangement or breakage of some part of the controlling mechanism may occur.

To allow the discharge valves to adjust themselves automatically to the varying conditions they must have some degree of freedom as to their time of opening. Only a partial control is practicable for any service in which the consumption of air is variable. Moreover, Corliss valves as used for compressors do not serve well for discharge valves where the ratio of compression is greater than, say, three or three and one-half; because they must then be set to open too late in the stroke to permit a free discharge. This applies to ordinary single-stage compressors, as well as to the high-pressure cylinders of two-stage machines. A number of devices have been introduced for dealing with these conditions; for example, relief valves working in conjunction with mechanically operated discharge valves.

Valve Motion of Norwalk Compressor. An adaptation of the Corliss valve gear is used for the low-pressure cylinder of this compressor (Fig. 73). One inlet and one discharge valve are set in chests at each end of the cylinder. Poppet valves are employed for high-pressure cylinder (as shown in Fig. 6, together with cross-sections of the low-pressure Corliss valves). In Fig. 73, the main valve-rod *a* is driven by a drag- or return-crank *b*, mounted on the fly-wheel crank-pin. The rod is pin-connected

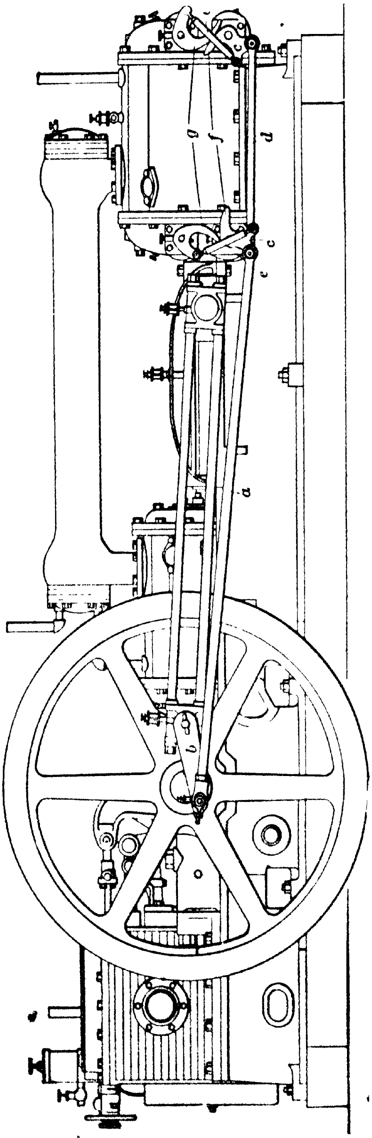


FIG. 73.—Valve Motion of Low-Pressure Air Cylinder. Norwalk Compressor.

to a short lever *c*, on the spindle of the forward inlet valve, and from this lever a link *d* passes to a corresponding connection with the inlet valve at the other end of the cylinder, the parts being so adjusted that one valve opens as the other closes. A positive movement of the valves is thus obtained. Cams *f f* and *g g*, for operating the discharge valves, are mounted on the respective inlet and discharge valve spindles, and form part of the short levers *c*. As each inlet valve oscillates, its cam rolls upon that of the discharge valve above it, each pair of cams being shaped so that the discharge valve is opened full when the cylinder pressure equals that in the discharge passage outside. Then, at the end of the stroke, when the cams move in the opposite direction, still rolling upon each other, the discharge valve is closed without shock by the connecting link *e*. This link is made of two telescoping parts, on the principle of a dash-pot, thus allowing freedom of movement for dealing with variable receiver pressure.

A number of other compressor builders have adopted modifications of the Corliss valve gear for the air cylinders.

Nordberg Valve Motion. For single-stage compressors the inlet valves are of the Corliss type, poppets being used for discharge (see Fig. 42). The inlet valves are operated positively from a triple wrist-arm, on the side of the air cylinder, driven by an eccentric on the crank-shaft. Connecting links pass from the wrist-arm to the valve levers. The lap of the valves can be altered by slightly shifting the angular position of the lever with respect to the valve spindle on which it is mounted. This is done by adjusting screws on the hub of the valve lever, the correctness of the valve motion being unaffected.

The stage compressors have similar inlet valves, a modified Corliss valve, with double ports, being used for discharge (Fig. 74). This valve is shown open on the right and closed on the left-hand end of cylinder. An opening in the center of the valve allows the air to discharge on both sides. In the axis of each Corliss valve are set a series of spring poppets, which act as relief valves when the receiver pressure falls below the normal. As shown by the cut the water-jacketed area on the cylinders is unusually

large, jackets being applied to the heads and around the valves, as well as on the cylinder barrels.

Another form of Nordberg valve gear is used in compressors intended for constant speed, either steam- or belt-driven. (Figs. 107, 108.) While the general construction of the air cylinders is the same as shown in Fig. 42, the inlet valves have a releasing mechanism. When the air pressure is normal the valves are operated as usual by wrist-plate links. But when the pressure increases the inlet valves are released and held open until the pressure drops; that is, the compressor is unloaded for the time

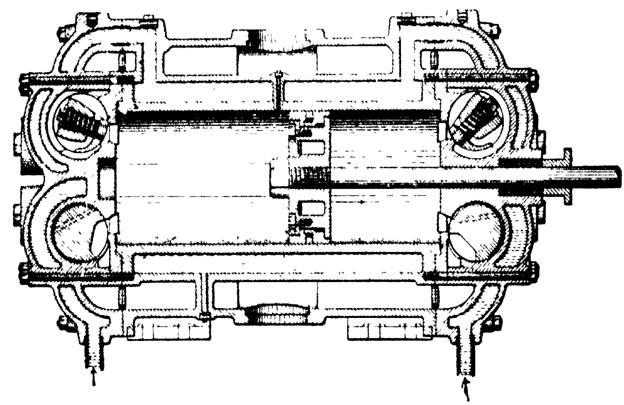


FIG. 74.—Air Cylinder of Nordberg Stage Compressor.

being, useful work ceasing. The release is effected by knock-off cams, similar to those used for Corliss steam valves, these cams being operated by a loaded plunger to which the compressed air is admitted when the pressure exceeds the normal. The compressor is thus self-regulating within small limits. For duplex compressors, added delicacy of regulation is obtained by designing the knock-off cams to unload in four successive steps, according to the variation in air pressure.

Laidlaw-Dunn-Gordon Valve Motions. Several types of mechanical valve motion were formerly made by these builders. A number are still in operation, but the makers now employ the

"feather valves," shown in Figs. 61, 62 and 63, Chap. VII. In the "Cincinnati" valve gear (Fig. 75) a single Corliss valve, at each end of the cylinder, serves as both inlet and discharge. The valve at the right-hand end of the cylinder is in position for admitting air, while that at the left is open for discharge, the corresponding inlet being closed. A large poppet is set vertically just above each Corliss valve. The latter is timed to open the port early enough in the stroke to leave the poppet free to rise whenever the cylinder pressure exceeds receiver pressure. At

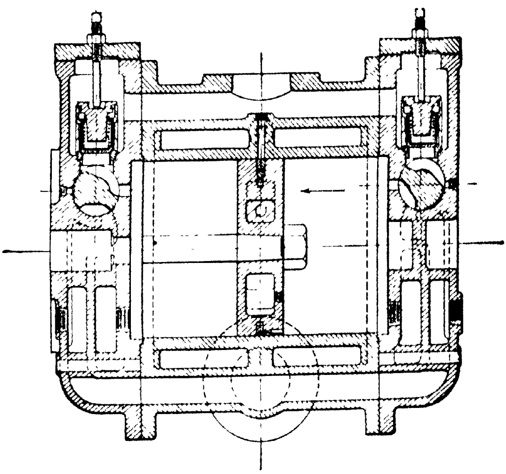


FIG. 75.—"Cincinnati" Valve Gear, Laidlaw-Dunn-Gordon Compressor.

the end of the stroke the Corliss valve takes its inlet position (right hand of cut), and at the same time, by shutting off the discharge, confines a little compressed air in the passage under the poppet. This air acts as a cushion, allowing the poppet to seat itself slowly and without shock, during the return stroke of the piston. Fixed mechanical control is thus exerted at three points: opening and closing of the inlet, and closing of the discharge.

Allis-Chalmers Valve Motions. Fig. 76 shows Corliss inlet valves, operated from a triple wrist-arm driven by an eccentric on the fly-wheel shaft.

The discharge valves (five in number for ordinary sizes of compressor) are spring-poppets of the cup form. Another design of discharge valve, formerly employed by these builders, consists of a cup poppet, without a spring, which is permitted to open freely, but is closed positively by a plunger actuated from a separate wrist-plate and eccentric. A single valve is placed in each cylinder head. The plunger, carried by exterior guides, works within the valve and is so timed that it forces the valve to its seat just at the end of the stroke. On

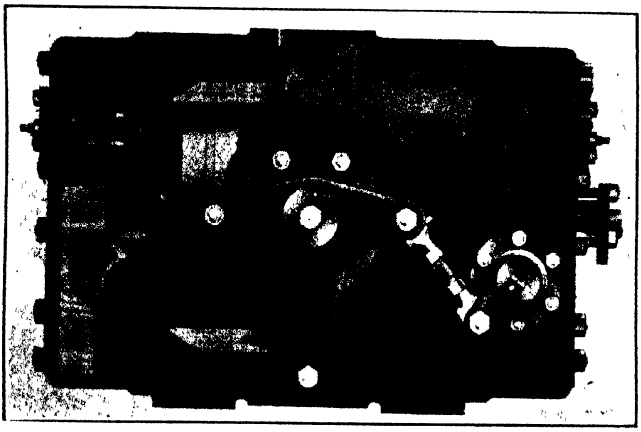


FIG. 76.—Standard Air Valve Motion. Allis-Chalmers Co.

the return stroke the plunger recedes, while the valve is held on its seat by the receiver pressure until the cylinder pressure rises sufficiently to open it. In closing the valve, the plunger is cushioned on the air in the cup valve, which is thus seated without shock.

Another design uses Corliss valves for both inlet and discharge. The time of closing the discharge is adjusted for the maximum working pressure. To allow for variations, small auxiliary spring poppets act as relief valves. These open freely when the receiver pressure falls below the pressure for which the Corliss valves are set.

In a later design (Figs. 14 and 43), 9 low-lift plate valves are used for discharge at each end of the cylinder. Light springs seat the valves promptly at the beginning of the intake stroke. No lubrication is required.

Sullivan Valve Motions (see Figs. 5, 7 and 8, Chap. II). In the straight-line, compound, two-stage compressors, Class "WC" (Figs. 7 and 8), the low-pressure cylinder has cup poppet inlets and Corliss discharge valves; the high-pressure

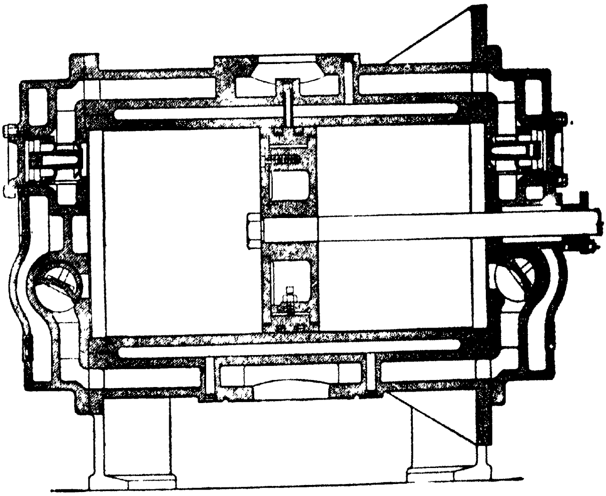


FIG. 77.—Sullivan Air Cylinder, Corliss Inlet and Poppet Discharge Valves.

cylinder has Corliss valves for inlet and poppets for discharge. The two-stage "WB-2" Class (Fig. 5), has a similar valve motion. The Corliss valves are operated from an eccentric pin (drag-crank) on the main crank-pin. Fig. 77 shows a design for a single-stage compressor. In an older pattern of these makers, the discharge valves are set radially around the upper part of the cylinder-head castings. The piston clearance can thus be made very small, and the valves, placed in removable seats, are surrounded by the water-jackets.

Other Mechanically Controlled Valve Motions are employed in a number of compressors, as the Franklin, Clayton, Rix, American, etc. With one exception, all of the mechanically controlled air valves referred to in this chapter are adaptations of the Corliss steam valve. A wholly different type, however, is the

Riedler Valve Motion. This has undergone radical modifications since its introduction, about 1888. Figs. 78, 79 and 80 show the most recent design. Though no longer made in this country, some Riedler compressors are still in use here, and the valve motion is an interesting one. Mechanical control is exerted through a wrist-plate A, operated by an eccentric on the fly-wheel shaft. Back of A is a sliding plate B, to which the links EE are pinned. Plate B is caused to reciprocate, through a distance equal to the lift of the valves, by two cams cut on the periphery of the wrist-plate and working against studs set in B. The motion thus transmitted through links E oscillates the rock-shafts D, and throws the forked levers C (Fig. 79), which control the closure of the valves.

The four valves, two suction and two delivery, are similar in design. They are double-seated poppets, the air passing within the seating ring as well as around its periphery. The valve stem passes through a stuffing-box in the cylinder head into a bonnet bolted on outside. Within the bonnet is a dash-pot, the piston of which is attached to the valve stem.

The inlet valve F, Fig. 79, operates as follows: At the beginning of the stroke the forked lever C is depressed by the rock-shaft and link, but leaves the valve free to open, its movement being steadied by the dash-pot piston G. The resistance presented by this piston is regulated by the screw H. For ordinary sizes of compressor the lift of the valve is 1 in., giving a large area of opening. (The 10¼-in. valve shown in the cut, which is for the low-pressure cylinder of a 24-in. and 38-in. by 48-in. compressor, has an area of 45 sq. ins.) Toward the end of the stroke the forked lever brings the valve gradually nearer its seat, as the piston velocity decreases. In completing its movement at the end of the stroke, the lever forces the valve

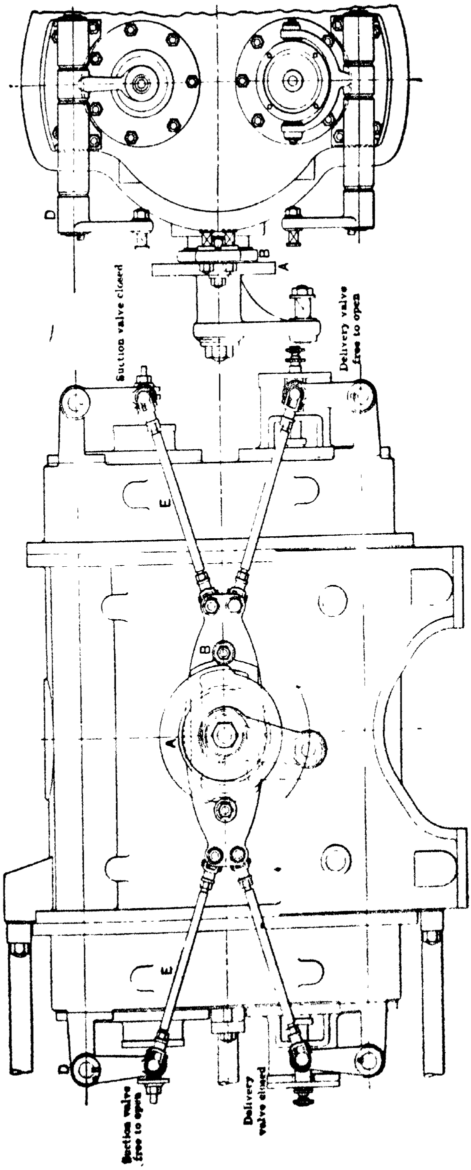


FIG. 78.—Riedler Air-Valve Motion, Low-Pressure Cylinder.

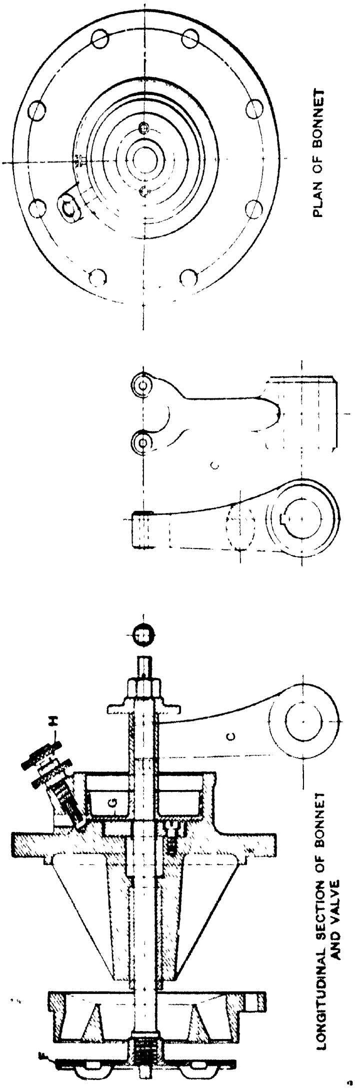


FIG. 79.—Details of Riedler Inlet Valve (Fig. 73).

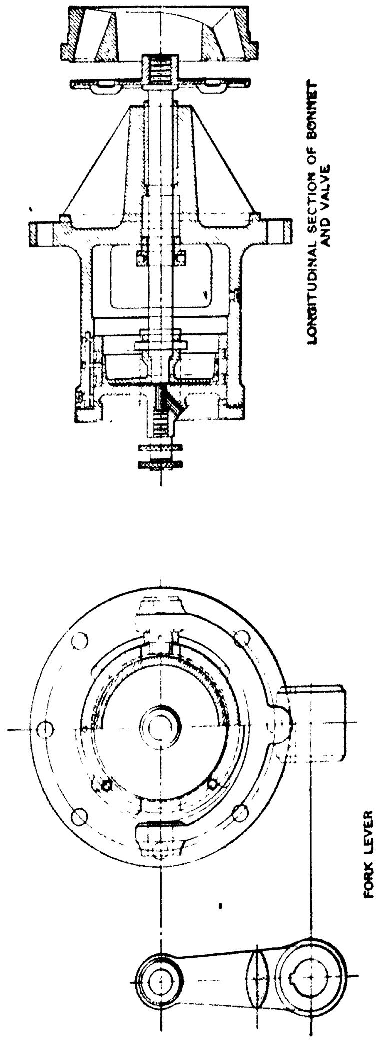


Fig. 80.—Details of Riedler Discharge Valve (Fig. 78).

upon its seat promptly. The valve thus attains its maximum lift in the middle of the stroke, when the velocity of the inflowing air is greatest, and is brought nearer its seat as the flow diminishes, so that closure is completed instantaneously at the proper time.

The control of the delivery valve is similar, though the details of its bonnet, dash-pot, and forked lever are different (Fig. 80).* At the proper point of stroke the lever is depressed, so that the valve is free to open when the air pressure in front of the advancing piston reaches receiver pressure. Then, as the velocity of outflow diminishes toward the end of the stroke, the valve approaches its seat, and closes promptly the instant the stroke reverses.

Köster valve is of the piston type. It is employed by Bailey & Co., Manchester, England, and several compressor builders on the continent of Europe. The valves, both inlet and discharge, are large in area and mounted on a longitudinal spindle deriving its reciprocating motion from an eccentric on the crank-shaft. The opening and closure of the inlet valves are positive, but the delivery port is opened by an independent poppet, encircling the spindle and provided with a light spring.

* The delivery valve in the cut is the same size as the inlet valve in Fig. 79, but is designed for a smaller compressor, 15″ and 24″×36″.

CHAPTER X

PERFORMANCE OF AIR COMPRESSORS *

THE duty of air compressors may be designated in three ways:

First. A useful standard of rating for ordinary purposes is the volumetric output, in terms of cubic feet of free air compressed per minute to a given pressure. The theoretical output is found by multiplying the net piston area in square feet by the distance travelled by the piston in feet per minute. The actual output is less than this on account of losses due to leaks, clearance, induction of warm air, friction of inlet valves, etc. In a well-designed compressor an allowance of 8 to 12% will cover these losses, which must not be confounded with the work required to overcome the friction of the compressor, and the added work due to the heating of the air while being compressed. The work losses are dealt with later.

Having found the volumetric capacity of the compressor, the volume V′ of this air, at any given pressure P′, is calculated by the formula: $V' = \frac{VP}{P'}$ where

V = initial volume of free air, cu.ft.;

P = normal absolute pressure of atmosphere (14.7 lbs.);

P' = terminal absolute pressure = gage pressure + 14.7 lbs.

For example, 100 cu.ft. of free air, compressed isothermally to 65 lbs. gage, will occupy a volume:

$$V' = \frac{100 \times 14.7}{65 + 14.7} = 18.45 \text{ cu.ft.}$$

Conversely, the volume of free air corresponding to 18.45 cu.ft. of air at 65 lbs. gage pressure is:

$$V = \frac{V'P'}{P} = \frac{18.45(65 + 14.7)}{14.7} = 100 \text{ cu.ft.}$$

* The deductions of the work formulas used here are given in Chapter III.

By applying the 8–12% allowance for losses stated above, sufficiently accurate results are obtained for practical purposes. As the volumetric output of a given size of cylinder depends on the density of the intake air, it will obviously be reduced when working at an altitude above sea-level (Chap. XIII).

Second. The size of the compressor may be designated in terms of the horse-power developed by the steam end, indicator cards being taken while running at normal working speed and while the usual volume of air is being compressed.

Third. The effective horse-power represented by the quantity of compressed air delivered is determined from an indicator card taken from the air cylinder. In testing a compressor it is customary to take a series of cards, simultaneously from both ends of the steam and air cylinders. They may then be compared, as shown by Fig. 15, Chap. II.

If indicator cards are not available, the theoretical horse-power for single-stage adiabatic compression may be calculated by the formula:

$$\text{H.P.} = \frac{144\text{PV}n}{33{,}000(n-1)}\left[\left(\frac{\text{P}'}{\text{P}}\right)^{\frac{n-1}{n}} - 1\right], \text{ in which}$$

P = normal atmospheric pressure, 14.7 lbs. per sq. in.;
P′ = final absolute pressure, lbs. per sq. in.;
V = the volume of free air compressed per min., cu.ft.;
n = exponent of the compression curve. For adiabatic compression, $n = 1.406$, and varies down to 1.18 or 1.2, according to the efficiency of the cooling arrangements, and whether single or stage compression For the best single-stage compressors, $n = 1.3$ (approx.).

For isothermal compression:

$$\text{H.P.} = \frac{144}{33\ 000} \times \text{PV}\left(\text{Nap. log } \frac{\text{P}'}{\text{P}}\right)^*$$

Table V shows the horse-powers required, under the conditions named, to compress one cubic foot of free air per minute:

* The Naperian or hyperbolic logarithm of a number is equal to the common logarithm multiplied by the constant 2.302585.

TABLE V.—SINGLE-STAGE COMPRESSORS

Gage Pressure, Lbs	Atmospheres Absolute, or Ratio of Compression $= \frac{P'}{P}$.	Single-Stage Compression, from Atmospheric Pressure at Sea-Level. Initial Temp., 60° F. Horse-power Required to Compress 1 Cu. ft. of Free Air.			
		Theoretical Horse-Power.		Actual Horse-Power (Approx.)	
		Isothermal Compression	Adiabatic Compression	Allowance for Losses above Adiabatic Compression, 10%	Allowance for Losses above Adiabatic Compression, 15%.
20	2.36	0551	0626	0689	0720
25	2.71	0637	0741	0815	0852
30	3.04	0713	0843	0927	0970
35	3.38	0782	0941	1035	1082
40	3.72	0842	1029	1132	1183
45	4.06	0895	1115	1226	1282
50	4.40	0950	1191	1310	1370
55	4.74	0994	1269	1396	1460
60	5.08	1041	1337	1471	1537
65	5.42	1081	1401	1541	1610
70	5.76	1123	1468	1615	1690
75	6.10	1162	1535	1688	1765
80	6.44	1195	1591	1750	1830
85	6.78	1224	1651	1816	1900
90	7.12	1256	1703	1873	1955
95	7.46	1287	1760	1936	2024
100	7.80	1315	1807	1988	2080
110	8.48	1366	1894	2083	2180
125	9.50	1442	2025	2227	2328

In columns 3 and 4 of Table V are the theoretical horse-powers required for isothermal and adiabatic compression. The results of isothermal compression are wholly unattainable in practice, and are placed here only for comparison. The figures in column 4 are based on the assumptions that there is no radiation of heat from the air cylinder, and that the temperature of the air after delivery has become normal, its volume being therefore reduced to that which is practically available for use. These figures include no losses except those due to the heating of the air while being compressed. But the full amount of loss represented by adiabatic compression can never be suffered in the operation of compressors, however imperfect their design. The actual compression line is always lower than the adiabatic

line, because of the radiation of heat through the cylinder walls. Even in single-stage compressors, properly water-jacketed and run at a reasonable piston speed, the compression line may fall considerably below the adiabatic. Whatever diminution of loss is effected by cooling the air in the cylinder may therefore be credited against the other unavoidable losses, partially offsetting them, *viz.:* frictional or mechanical loss in the compressor, friction of inlet valves, heating of the intake air by contact with the hot metal surfaces, and piston clearance.

In the absence of indicator cards, estimates based on practice may be made of the compressor horse-power. In columns 5 and 6, of Table V, are shown the actual horse-powers required to compress 1 cu.ft. of free air, under the conditions stated at top of the columns. Thus, in column 5, 10% is assumed as a fair estimate, in case of large, well-designed and operated single-stage compressors in good running order, of the additional power required, over and above that for adiabatic compression. This 10% is taken as the algebraic sum of: the loss in purely adiabatic compression, minus the effect of ordinary water-jacket cooling, plus the four losses mentioned at end of preceding paragraph. In column 6, the power consumed in adiabatic compression is increased by 15%, which represents relatively poorer work. (See also Table VI).*

It must be remembered that, if the compressor is small, or in poor condition, or is run at too high a speed, the required horse-power is greater. In such cases, the added percentages in column 6 of Table V and columns 5 and 7 of Table VI may be increased to 18 or 20%.

The figures in columns 3 and 4 or 5 and 6 of Table V (which are for *free* air), if multiplied by the corresponding ratios of compression (column 2), will give the respective theoretical and actual power costs of furnishing 1 cu.ft. of *compressed* air, at the gage pressures stated.

* Since the previous edition of this book was published, the working efficiency of the better types of compressor has improved. In view of this, the figures in Tables V and VI, showing the actual horse-power per cu.ft. of free air compressed, have been materially reduced.

Work of Stage Compressors (see Equations 19 and 20, Chap. III). The theoretical horse-power for two-stage compression is:

$$\text{H.P.} = \frac{2 \times 144}{33{,}000} \times \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{2n}} - 1\right]$$

For three-stage compression:

$$\text{H.P.} = \frac{3 \times 144}{33{,}000} \times \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{3n}} - 1\right]$$

Reducing the constants, and for a volume of 1 cu.ft. free air:

$$\text{Two-stage,}\quad \text{H.P.} = 0.449\left[\left(\frac{P'}{P}\right)^{0.144} - 1\right]$$

$$\text{Three-stage,}\quad \text{H.P.} = 0.6735\left[\left(\frac{P'}{P}\right)^{0.0962} - 1\right]$$

TABLE VI.—TWO- AND THREE-STAGE COMPRESSORS

Gage Pressure, Lbs.	Ratio of Compression $= \frac{P'}{P}$	Horse-Power per Cu. Ft. of Free Air at Sea-Level.				
		Isothermal Compression	Two-Stage Compression		Three-Stage Compression	
			Adiabatic Compression	Actual H.P. on basis of Adiabatic Comp'n +15%	Adiabatic Compression	Actual H.P. on basis of Adiabatic Comp'n +12%
70	5.76	0.1123	0.129	0.148		
80	6.4	.1195	.138	.159		
90	7.12	.1256	.147	.169		
100	7.80	.1315	.154	.177	0.145	0.162
120	9.16	.1420	.169	.194	.158	.177
140	10.50	.1508	.181	.208	.169	.189
160	11.88	.1583	.192	.221	.179	.200
180	13.24	.1654	.202	.232	.188	.210
200	14.60	.1720	.212	.244	.196	.219
250	18.00	.1853	.231	.266	.213	.238
300	21.40	.1963	.249	.285	.228	.255
350	24.80	.2058	.264	.303	.241	.270
400	28.20	.2140	.277	.318	.252	.282
450	31.62	.2215	.289	.332	.262	.293
500	35.01	.2280			.271	.303
550	38.41	.2339			.280	.313
600	41.80	.2393			.288	.322
650	45.21	.2443			.295	.330
700	48.62	.2490			.301	.337
800	55.42	.2574			.314	.352

Tables V and VI may be used for any altitude above sea-level, by considering the ratio of compression in column 2. Thus, to find the horse-power for two-stage compression per cu-ft. free air at 10,000 ft. altitude and 80 lbs. gage: Ratio of compression

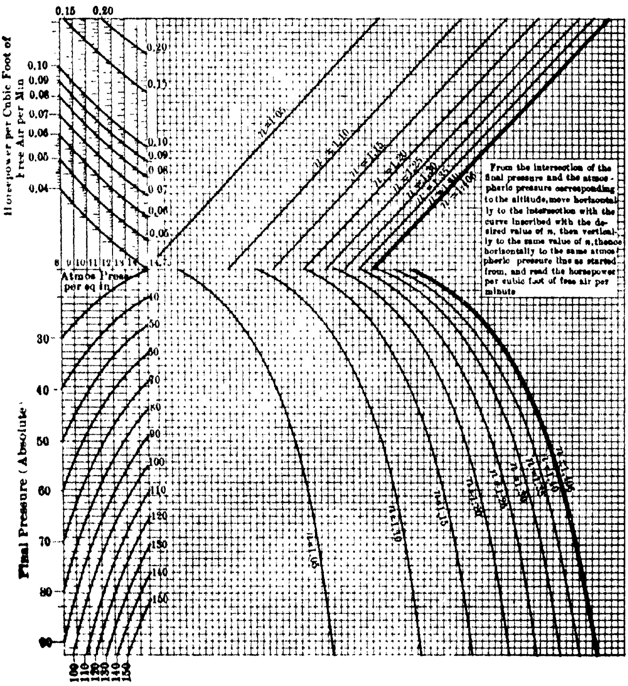


FIG. 81. Single-stage Compression.

$= \dfrac{80+10.07}{10.07} = 8.94$ (the factor 10.07 being the barometric pressure at 10,000 ft., from Table XIII, Chap. XIII). From Table VI, column 2, 8.94 compressions (by interpolation) correspond to a gage pressure of about 118 lbs. at sea-level, the horse-power for which (column 5) equals 0.191 per cu.ft. of free air. At sea-level, the horse-power would be 0.159.

Graphic Determination of the horse-power required to compress 1 cu.ft. of free air to a given pressure (C. W. Crispell, *Trans. Am. Inst. Min. Engs.*, June, 1917). For this the nomograms in Figs. 81 and 81a are more convenient than the formulas. The values of P (atmos. pressure) in lbs. per sq. in., for different

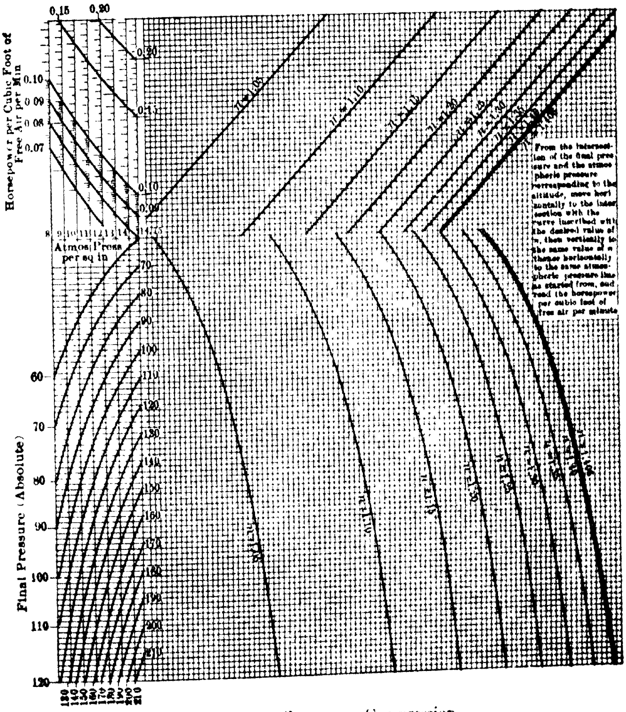


FIG. 81a. Two-stage Compression

altitudes, are given in Table XIII, column 3 (Chap. XIII). Directions for using the charts are printed on them. Note that the scale for the H.P. per cu.ft. of air begins at a different ordinate for each value of P. By a series of examples in the paper quoted, it is shown that the nomograms usually give results from 0.7% to 1.6% higher than those from the formulas.

Table VII is useful for calculations based on volumes and mean cylinder pressures. The mean pressures per stroke (columns 5 and 6) are obtained from the formulas for isothermal and adiabatic single-stage compression, by making $V=1$, thus:

$$\text{Mean pressure per stroke (isothermal)} = P \times \text{Nap. log}\,\frac{P'}{P}$$

$$\text{Mean pressure per stroke (adiabatic)} = 3.463P\left[\left(\frac{P'}{P}\right)^{0.29} - 1\right]^*$$

The work done during one stroke is equal to the mean pressure multiplied by the volume in cu.ft. traversed by the piston.

When air is compressed adiabatically, the relation between its temperature T, at the beginning of compression, and the terminal temperature T′, is shown by:

$$\frac{T'}{T} = \left(\frac{V}{V'}\right)^{n-1}, \text{ whence } T' = T\left(\frac{V}{V'}\right)^{n-1}$$

The final temperature may also be found from the formula:

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}$$

T and T′ being absolute temperatures and P, P′ absolute pressures.

The compression curve of an air-indicator card may be constructed as follows, PV being the pressure and volume at one point of the curve and P′V′ the pressure and volume corresponding to any other point. Designating the index number of the curve by x:

$$\frac{P}{P'} = \left(\frac{V'}{V}\right)^x. \text{ From this,}$$

$$\log\left(\frac{P}{P'}\right) = x \log\left(\frac{V'}{V}\right); \text{ wnence, } x = \frac{\log\left(\frac{P}{P'}\right)}{\log\left(\frac{V'}{V}\right)}$$

* The constant $3.463 = \frac{n}{n-1} = \frac{1.406}{.406}$; exponent $0.29 = \frac{n-1}{n} = \frac{1.406-1}{1.406}$.

TABLE VII *

Gage Pressures.	Atmospheres.	Volume with Air at Constant Temperature (Isothermal).	Volume with Air Not Cooled (Adiabatic).	Mean Pressure per Stroke, Air at Constant Temperature. Lbs.	Mean Pressure per Stroke, Air Not Cooled. Lbs.	Temperature of Air; Not Cooled. Deg. F.
0	1	1	1	0	0	60°
1	1 068	9363	9500	96	975	71
2	1 136	8803	9100	1 87	1 91	80 4
3	1 204	8305	8760	2 72	2 80	88 9
4	1 272	7861	8400	3 53	3 67	98
5	1 340	7462	8100	4 30	4 50	106
10	1 680	5952	6900	7 62	8 27	145
15	2 020	4950	6060	10 33	11 51	178
20	2 360	4237	5430	12 62	14 40	207
25	2 700	3703	4940	14 59	17 01	234
30	3 040	3289	4538	16 34	19 40	252
35	3 381	2957	4200	17 92	21 60	281
40	3 721	2687	3930	19 32	23 66	302
45	4 061	2462	3700	20 57	25 59	321
50	4 401	2272	3500	21 69	27 39	339
55	4 741	2109	3310	22 76	29 11	357
60	5 081	1968	3144	23 78	30 75	375
65	5 423	1844	3010	24 75	32 32	389
70	5 762	1735	2880	25 67	33 83	405
75	6 102	1639	2760	26 55	35 27	420
80	6 442	1552	2670	27 38	36 64	432
85	6 782	1474	2566	28 16	37 94	447
90	7 122	1404	2480	28 89	39 18	459
95	7.462	1340	2400	29 57	40 40	472
100	7 802	1281	2320	30 21	41 60	485
105	8 142	1228	2254	30 81	42 78	496
110	8 483	1178	2189	31 39	43 91	507
115	8 823	1133	2129	31 98	44 98	518
120	9 163	1091	2073	32 54	46 04	529
125	9 503	1052	2020	33 07	47 06	540
130	9.843	1015	1969	33 57	48 10	550
135	10.183	0981	1922	34 05	49 19	560
140	10.523	0950	1878	34 57	50 02	570
145	10.864	0921	1837	35 09	51 00	580
150	11.204	0892	1796	35 48	51 89	589
160	11 880	0841	1722	36 29	53 65	607
170	12 560	0796	.1657	37 20	55 39	624
180	13 240	0755	1595	37 96	57 01	640
190	13 920	.0718	1540	38 68	58 57	657
200	14 600	0685	1490	39 42	60 14	672

* Kents' "Mechanical Engineers' Pocket Book." Taken from a table in Richards' "Compressed Air," p. 20.

The several lines of an air-card have significations entirely different from those of a steam card. In Fig. 82 (an ideal card), AB is the admission line, BC the compression line, CD the delivery or discharge line, and DA the re-expansion line. DA represents the effect of the re-expansion of the clearance air, on beginning a stroke. Comparing the lines of the air and steam cards, they are found to be reversed:

AIR CARD.	STEAM CARD.
Admission line	Back pressure or exhaust line.
Compression line	Expansion line
Delivery line	Admission line
Re-expansion line	Compression line

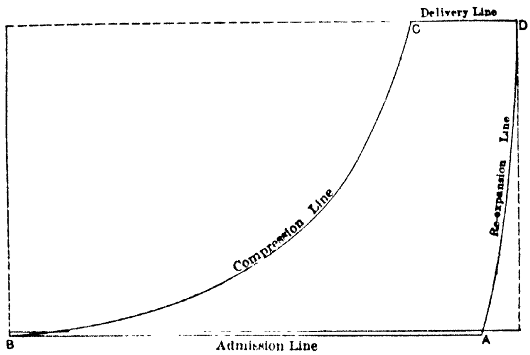


FIG. 82

The elements of an air card, together with the work done, as represented by the several lines and areas, are further elucidated by Fig. 82a, the compression being adiabatic.

Let AD = normal atmospheric line at sea-level;

AG = P = corresponding atmospheric pressure, acting behind the piston at the beginning of the stroke (neglecting valve resistances and effect of clearance of previous stroke);

GE = AD = length of stroke of piston;

AB = adiabatic compression curve;

BC = delivery line.

At the point B the useful work of compression ceases; during the remainder of the stroke the volume of compressed air V′, at the absolute pressure P′, is being forced out of the cylinder through the delivery valves.

The area ABFG = the absolute work of compression.

The area BCEF = the absolute work of delivery.

The sum of these areas represents the total absolute work (that is, on the basis of absolute pressure) done during compression and delivery.

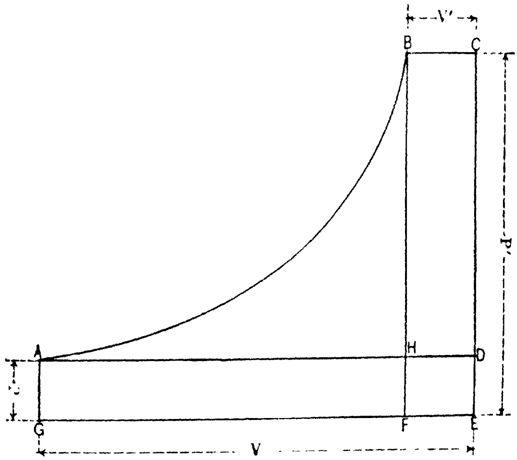


FIG. 82a

Area ADEG = work done during the entire stroke by atmospheric pressure behind the piston.

Area ABH = net work of compression.

Area BCDH = net work of delivery.

Area ABCDA = total net work for entire stroke.

From this analysis another method may be derived for calculating the theoretical horse-power required for compressing air. It will be found useful, when a table of temperatures of compression is available.

Let w = weight of 1 cu.ft. of free air = .0765 lb.;
C_p = specific heat of air at constant pressure = .2375;
C_v = specific heat of air at constant volume = .1689.

$$\text{Whence, } \frac{C_p}{C_v} = n = 1.406, \text{ and } \frac{n}{n-1} = 3.463.$$

J = Joule's heat unit, taken as 778 ft.-lbs.

The work represented by the area ABH =

$$J \times w \times C_v(T' - T) - P(V - V').$$

Also, the work done during delivery = BCDH = $V'(P' - P)$. Hence, the total net work for one stroke of the piston

$$= \text{area ABCDA} = J \times w \times C_v(T' - T) - (PV - P'V').$$

If C_p be substituted for C_v, then $PV = P'V'$, according to the general equation for air compression, and the total work

$$W = J \times w \times C_p(T' - T).$$

Substituting for J, w, and C_p, their constant numerical values:

$$W = 14.13(T' - T),$$

or, to compress 1 cu.ft. of air per min., at 60° F., and at sea-level

$$\text{H.P.} = 0.225\left[\frac{T'}{T} - 1\right].$$

By referring to the last column of Table VII and remembering that T and T′ are absolute temperatures, *i.e.*, thermometric temperatures plus 459° F., the horse-power required for compressing 1 cu.ft. of free air adiabatically to any gage pressure may readily be calculated.

Other expressions for the mean effective pressures may also be deduced from what precedes. M.E.P. for the entire stroke =

$$P\frac{n}{n-1}\left(\frac{T'}{T} - 1\right) = 3.46P\left(\frac{T'}{T} - 1\right) = 3.46P\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\text{M.E.P. during delivery} = \frac{V'}{V}(P' - P),$$

The M.E.P. for compression only is found by taking the difference between the pressures calculated by the last two formulas.

The results from the above expressions for work and mean effective pressure are theoretical. To find the actual horsepower required, allowances must be made for the losses experienced in the operation of the compressor, as already set forth.

Compressor Tests. To indicate the observations required to secure the data for the complete test of a compressor, together with the deductions from the observed data, the following record of the test of a compound, two-stage Nordberg compressor, at the mines of the Tennessee Copper Co., will be found useful.* It will be noted that items 28, 29, and 32 to 35, were necessary in this case, because the boiler plant supplied steam for a hoisting engine and an independent condenser, as well as for the compressor. Though the hoist was not running, steam was passing continuously to the jackets of the cylinders. The same conditions would often be met in other tests. The boiler feed water was taken from a wooden tank, and during the run this water was supplied from two barrels on scales set temporarily over the tank. The water of condensation from steam jackets and reheater was drawn off continuously and also weighed. The calorimeter tests were made with a Peabody throttling calorimeter. Eight sets of indicator cards were taken during the 8-hour test, at hourly intervals.

ITEMS OF COMPRESSOR TEST (Altitude, 1,800 feet)

1.	Date of test, February 16, 1902	
2	Duration of test, hours	8
3.	Diameter of high-pressure steam cylinder (steam jacketed), inches.	14
4.	Diameter of low-pressure steam cylinder (steam jacketed), inches	28
5.	Diameter of low-pressure air cylinder, inches.	$24\frac{1}{2}$
6.	Diameter of high-pressure air cylinder, inches.	$15\frac{1}{4}$
7.	Stroke of all pistons, inches	42
8.	Diameter of piston rods, inches	$2\frac{11}{16}$
9.	Revolutions of engine, average per minute.	90
10.	Piston speed per minute, feet	630

* Abstracted from an article by J Parke Channing, *Mines and Minerals*, May, 1905, p. 475.

11. Steam-gage pressure, average, pounds	145.9
12. Temperature of steam in steam pipe, average, degrees F.	364
13. Steam pressure in reheating receiver, average, pounds	8
14. Vacuum in condenser, average, inches	25.66
15. Air pressure in intercooler, average, pounds	22.63
16. Air pressure in receiver, average, pounds	79.3
17. Temperature of air at intake, average, degrees F.	65.0
18. Temperature of air leaving low-pressure cylinder, average, degrees F.	211.5
19. Temperature of air leaving intercooler, average, degrees F	78.5
20. Temperature of air leaving high-pressure cylinder, average, degrees F.	240.0
21. Indicated horse-power in high-pressure steam cylinder, average	140.12
22. Indicated horse-power in low-pressure steam cylinder, average	153.03
23. Indicated horse-power in both steam cylinders, average	293.15
24. Indicated horse-power in low-pressure air cylinder, average	143.79
25. Indicated horse-power in high-pressure air cylinder, average	135.02
26. Indicated horse-power in both air cylinders, average	278.81
27. Feed water weighed to boilers, pounds	43,343
28. Re-heater and jacket water from compressor, weighed, pounds	4,081
29. Average temperature of re-heater and jacket water, degrees F	356.7
30. Total heat in 1 pound of steam at 356.7 degrees F., heat units	1,190.7
31. Total heat in 1 pound of water at 356.7 degrees F., heat units	328.9
32. Equivalent credit for re-heater and jacket water, pounds	1,127.00
33. Water weighed from condensation in hoisting-engine jacket, pounds	1,781.00
34. Steam used to run condenser, pounds	4,300.00
35. Total credits to feed water, pounds	7,228.00
36. Total feed water charged to engine, pounds	36,115.00
37. Moisture in steam shown by Peabody calorimeter, per cent	1.30
38. Credit for moisture in steam, pounds	473.00
39. Total steam charged to engine, pounds	35,642.00
40. Dry steam per hour charged to engine, pounds	4,455.00
41. Steam consumption per indicated horse-power per hour, pounds	15.19
42. Guaranteed steam consumption per indicated horse-power per hour, at 92 revolutions per minute, pounds	14.00
43. Excess of steam consumption per indicated horse-power per hour over guarantee, pounds	1.19
44. Theoretical delivery of free air per minute at 90 revolutions, cubic feet	2,037.8
45. Slip of air (percentage)	3.0
46. Actual slip of air per minute, cubic feet	61.1
47. Actual delivery of free air per minute, average cubic feet	1,976.7
48. Theoretical horse-power required to compress and deliver actual delivery of air at receiver pressure by adiabatic compression	306.53
49. Theoretical horse-power required to compress and deliver actual delivery of air at receiver pressure by isothermal compression	229.00
50. Actual horse-power shown by air indicator cards	278.81
51. Actual horse-power shown by steam indicator cards	293.15
52. Actual horse-power consumed by friction of engine	14.34
53. Efficiency ratio between steam and air cylinders, per cent	95.1

54. Efficiency ratio between steam and air cylinders guaranteed by builder, per cent 87.0
55. Efficiency of steam, or ratio of steam indicated horse-power to theoretical air indicated horse-power, isothermal compression, per cent. 78.1

One of the combined indicator cards, from which the averages in items 21–26 were calculated, is shown in Fig. 83.

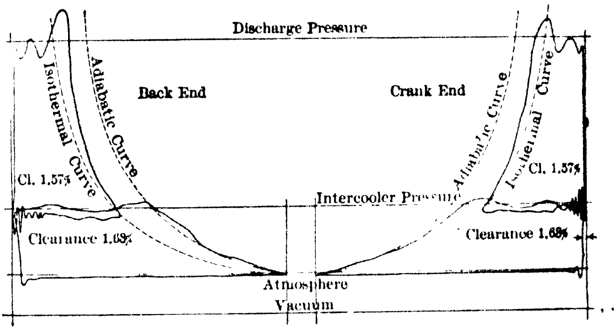


FIG. 83.—Combined Cards Two-Stage Nordberg Compressor.

In further illustration of the performance of air compressors, the combined card from an Ingersoll-Rand "Imperial Type 10" two-stage compressor, taken at one of the Berwind-White Coal and Coke Company's mines, is given in Fig. 84.

Figs. 85 and 86 show shop-test cards from the air cylinders of an Ingersoll-Rand style "O" compressor.

A Record of Field Tests. It would undoubtedly tend to secure greater economy in the production of compressed air, if superintendents and master mechanics gave more attention to the actual results produced by the operation of compressors in their charge, and study carefully the frequently unfavorable conditions under which these machines are called upon to work.

Few records of the actual effective horse-power of air compressors have been published. To express the efficiency, it is customary to divide the horse-power of the air cylinder by the horse-power of the steam cylinder, as determined by indicator cards. The manufacturer of air compressors usually rates his

machine on the basis of its mechanical efficiency, without taking into consideration any losses except those of friction. Such a criterion does not properly measure the relative commercial values of compressors, nor does it present any indication as to

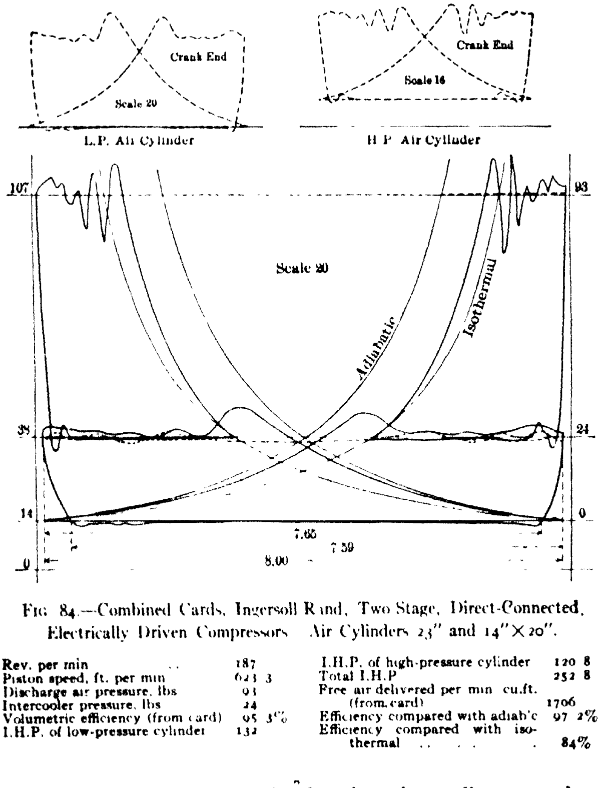


FIG. 84.—Combined Cards, Ingersoll Rand, Two Stage, Direct-Connected, Electrically Driven Compressors Air Cylinders 23" and 14" × 20".

Rev. per min ..	187	I.H.P. of high-pressure cylinder	120.8
Piston speed, ft. per min	623.3	Total I.H.P	252.8
Discharge air pressure, lbs	93	Free air delivered per min cu.ft. (from card)	1706
Intercooler pressure, lbs	24	Efficiency compared with adiab'c	97.2%
Volumetric efficiency (from card)	95.3%	Efficiency compared with isothermal	84%
I.H.P. of low-pressure cylinder	132		

the effective horse-power developed under ordinary working conditions.

A series of tests were made in 1909 by Richard L. Webb, consulting engineer, of Buffalo, N. Y., on a large number of compressors in a well-known Canadian mining district. In conducting these tests, Mr. Webb had access to plants which had been in

operation for a year or more under normal working conditions, and I believe his results will be of value not only to users of air compressors, but also to the manufacturers. As a rule, the plants tested were in the care of competent machinists and in good

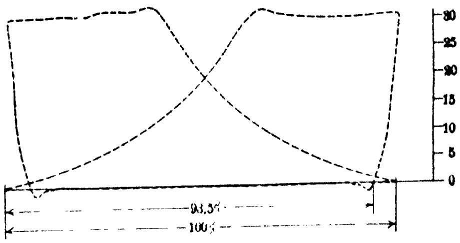


FIG. 85.—Card from $30\frac{1}{4}''\times24''$ L. P. Air Cylinder of Style "O," Ingersoll-Rand Compressor. (St. pressure, 115 lbs., air pressure, 28 lbs., r.p.m., 100; spring, 20.)

running order, so that the results obtained may be taken as representing a fair average of current practice in the United States and Canada. The results of a few of these tests are given here to show the importance of determining the actual

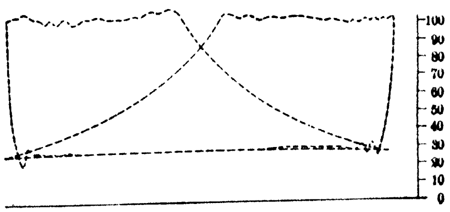


FIG. 86.—Card from $18\frac{1}{4}''\times24''$ H.P. Air Cylinder, Style "O," Ingersoll-Rand Compressor. (St. pressure, 115 lbs., air pressure, 100 lbs.; r.p.m., 100; spring, 60.)

efficiency of air compressors when working under the conditions prevailing in most mines.

Mode of Conducting the Tests. The following plan was employed in each case: *First*, a boiler test was run for not less than two weeks, the coal being carefully weighed, the boiler feed water measured, and the total revolutions of the compressor

recorded by a revolution counter. From these data, the cost per boiler horse-power and the average speed of the compressor were determined. *Second*, the compressor was operated at different speeds over its entire range. By means of a meter installed in the steam pipe near the throttle, the total steam consumed, in pounds per hour, was measured. Indicator cards were taken of all cylinders, together with temperatures at the air inlet, intercooler, and discharge. To measure the actual

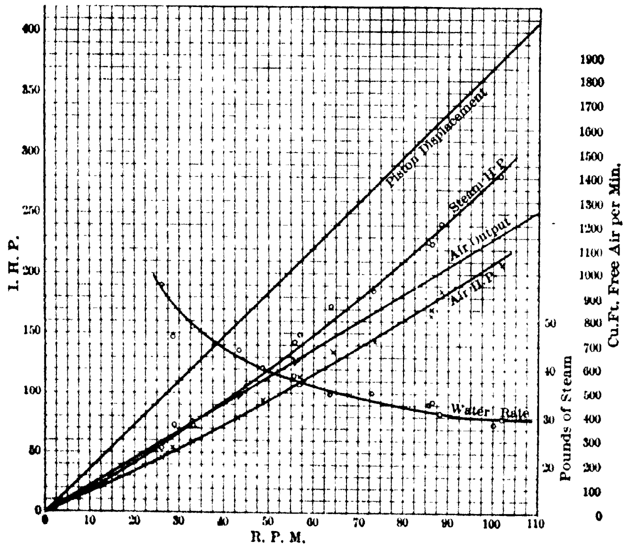


FIG. 87.—Compressor Plant No. 1.

volume of air delivered, a meter was placed in the discharge pipe outside of the receiver. A number of simultaneous readings on all instruments were taken at each speed. From these were calculated the total horse-power of the steam and air cylinders, the steam consumption, and the total piston displacement per minute.

The air and steam meters were of the Dodge type, as modified by the General Electric Company, and were operated by

their expert sent for this purpose. The indicators were of the Roberts-Thompson and the American-Thompson make, which are well known and generally accepted as standard. Their springs were calibrated by a standard gage.

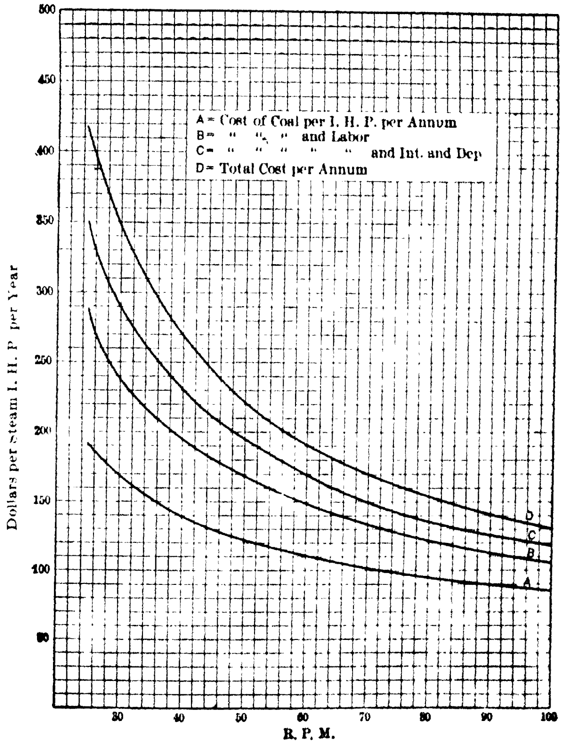


FIG. 88.—Compressor Plant No. 1.

Results of the Tests. As was to be expected, the friction loss was found to be only a small item in the total. The other losses, which are frequently overlooked or disregarded, played a large part in cutting down the efficiency. The capacity of air compressors is usually rated according to the volume of the cylinders.

On this basis, the mechanical efficiency only is given. For example, if the horse-power of the air cylinder is 100, and that of the steam cylinder 110, the efficiency of the compressor is rated as 91%. This rating disregards the losses due to adiabatic compression, heating of the cylinder and friction of the inlet and delivery valves. The tests show the loss of the engine itself to be usually not less than 10% and often considerably larger. Losses from the other causes mentioned ranged from 30% up.

As Mr. Webb is not at liberty to disclose the identity of the particular plants, each test has been designated by a number.

Test of Plant Number One. This consists of three 125 H.P. return tubular boilers (one being held in reserve), supplying steam for a cross-compound condensing air compressor of standard make. The steam cylinders have Meyer valve gear and are 16 in. and 28 in. diameter by 24 in. stroke. The two-stage air cylinders are 28 in. and 18 in. by 24 in. From a two weeks' run the following results were obtained.

Total coal burned, lbs	264,300
Total feed water, cu ft	37,459
Total feed water, lbs	2,335,568
Average temperature of feed water, degrees F. .	131
Average evaporation per lb coal consumed, lbs . .	8 72
Average revolutions per minute .	63 1
Indicated horse power of steam end, corresponding to 63.1 R P.M.	161
Corresponding indicated horse-power of air end.	123
Average steam pressure, lbs	115
Average vacuum, lbs	10 5
Average air pressure, lbs	96
Average temperature of outside air, degrees F	24
Average air piston displacement at 70° F., cu ft	1172
Average metered output corrected to 70° F, cu ft	758

The average evaporation, of 8.72 lbs. of water from 131° F. to an average steam pressure of 115 lbs., is equivalent to 9.83 lbs. of water evaporated from and at 212° F. per lb. coal consumed. At the average compressor speed of 63.1 rev. per min. the metered output was equivalent to 758 cu.ft. of free air per min., the piston displacement being 1,172 cu.ft. per min. Table VIII and Fig. 87 present the principal data of this test run.

To find the average operating results, the curves at 63 revolutions should be followed, at which the indicated horse-power of the steam cylinder was 160.8, and that of the air cylinder, 123, showing the mechanical efficiency to have been 76.5%.

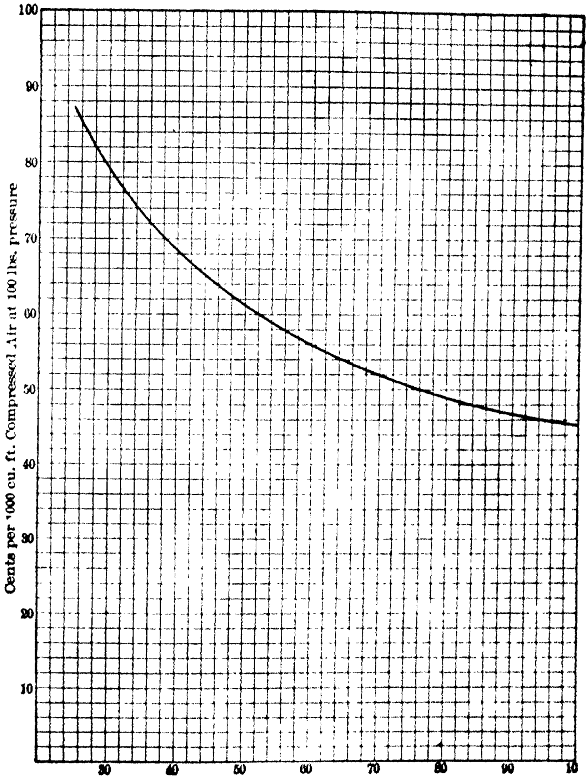


FIG. 89.—Compressor Plant No. 1.

The theoretical horse-power required to compress isothermally 1 cu.ft. of free air per min. to 96 lbs. (the average pressure) is 0.129. The theoretical useful work done by the compressor is, therefore, $758 \times .129$ or 97.8, and the net total efficiency of the compressor is $97.81 \div 161$ or 60.8%.

TABLE VIII.—TEST ON PLANT NO. 1

COMPOUND CONDENSING STEAM CYLINDERS, 16" AND 28", WITH MEYER VALVE GEAR, AIR CYLINDERS, 28" AND 18", STROKE 24"

Revolutions per Minute.	Steam Pressure, Pounds.	Air Pressure, Pounds.	High-Pressure Side				Low-Pressure Side				Total Steam, I.H.P.	Total Air, I.H.P.	Friction Loss	Friction Loss, Per Cent	Air Line Temperature	Cu. ft. Displacement, 70° F	Metered Air Output, Cu.Ft. per Min., 70° F	Calculated Output, Cu.Ft.
			Steam.		Air		Steam.		Air.									
			M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.								
25.8	125	75	55.0	34	33.95	20.68	10.3	19.75	14.31	21.68	53.75	42.36	11.39	21.2	220	480	340	295
32.7	126	87	63.2	49.5	39.65	30.65	10.7	26.0	14.69	28.2	75.5	58.85	16.65	22.05	245	610	415	375
43.0	126	95	63.5	65.45	42.44	43.10	11.65	37.2	14.16	35.73	102.7	78.8	23.9	23.2	250	800	510	492
55.5	129	100	64.6	86.0	46.15	60.4	13.01	56.2	15.32	49.0	142.2	110.3	31.9	22.4	258	1030	605	635
64.0	128	100	65.65	100.7	48.25	72.8	15.1	71.8	15.85	50.5	172.5	132.3	40.2	23.3	268	1190	650	730
73.0	126	94	61.2	107.1	42.88	73.8	14.78	80.2	15.64	67.0	187.3	140.8	46.5	24.8	202	1355	690	835
88.0	125	97	61.62	129.9	48.4	100.5	17.12	112.0	16.56	86.7	241.9	186.2	55.7	23.0	286	1655	740	898
101.08	127	95	63.85	155.9	47.75	114.7	16.5	124.9	15.8	94.5	280.8	209.2	71.6	25.5	280	1890	760	1165

NOTE—The above data are from a table of fourteen readings taken at different speeds

TABLE IX.—TEST ON PLANT NO. 1

POWER COSTS DUPLEX MEYER STEAM, COMPOUND AIR END OPERATED CONDENSING

Revolutions per Minute	Evaporation.	Water Rate.	Lbs. Coal per I.H.P. Hour	Steam I.H.P.	Pounds Coal per Hour.	Coal		Labor per I.H.P.	Coal and Labor per I.H.P.	Int. and Depreciation per I.H.P.	Coal, Labor, Int. and Depreciation per I.H.P.	Repairs, etc. per I.H.P.	Total Cost per I.H.P.	Cu. ft. Compressed Air per Hour.	Operating Hourly Costs.	Cents per 1,000 Cu.ft.
						Cost per Annum	Cost per I.H.P. per Annum									
25	8.08	59	7.30	55	401.5	$10,550	$192.00	$97.00	$289.00	$62.00	$351.00	$62.00	$414.00	2963	$2.59	.874
40	8.31	44	5.30	95	503.5	13,240	130.00	56.00	105.00	36.00	231.00	36.00	268.00	4238	2.90	.684
60	8.62	36.5	4.23	152	642.9	16,900	111.00	35.00	146.00	23.00	169.00	23.00	191.00	5963	3.32	.557
80	8.82	32	3.67	211	774.4	20,350	96.45	25.00	121.00	16.00	137.00	16.00	154.00	7650	3.72	.487
100	8.96	29.5	3.29	278	914.6	24,050	86.60	19.00	106.00	12.00	118.00	12.00	130.00	9308	4.14	.445

TABLE X.—TEST ON PLANT NO. 2

DUPLEX SIMPLE STEAM CYLINDERS, 14"×22", MEYER VALVE GEAR, TWO-STAGE AIR END, 22" AND 14"; 22" STROKE.
RATED CAPACITY, 1 050 CU FT PER MINUTE AT 105 R P M.

Revolutions per Minute.	Gage Pressure, Lbs		High-Pressure Side				Low-Pressure Side				Steam Total	Air Total	Per Cent Air of Steam	Pounds Steam, per I H P Hour	Metered Air Output Cu ft. per Minute, 70° F.	Displacement, Cu.ft. per Min., 70° F.
			Steam.		Air		Steam		Air							
	Steam	Air	M E P	I H P	M E P	I H P	M E P	I H P	M E P	I H P	I H P.	I H P				
14.3	90	100	44.62	11.3	43.8	10.7	43.2	10.55	14.4	8.71	21.85	19.4	88.9	78.9	295	151
46.0	90	95	45.5	37.0	41.1	32.4	43.3	34.0	15.75	30.6	71.0	63.0	88.8	46.32	564	484
51.0	85	98	45.4	39.6	41.0	30.3	43.9	38.2	15.06	33.4	77.3	69.7	90.3	53.9	654	535
82.0	85	55	42.2	59.1	25.0	35.0	32.5	45.5	15.8	54.6	104.6	90.5	86.5	42.4	744	864
103.0	77	93	48.9	86.1	40.1	70.5	44.1	78.2	17.03	74.0	104.3	144.5	88.0	43.1	1000	1083
111.0	82	75	46.7	88.6	32.3	61.4	40.5	79.9	10.8	78.5	165.5	130.9	84.6	38.9	1025	1161

NOTE.—The above data are from a table of readings taken at fourteen different speeds.

Table IX shows the actual cost of running this compressor at different speeds. The data were furnished by the owner and are based on one year's operation. In Fig. 88 these costs are

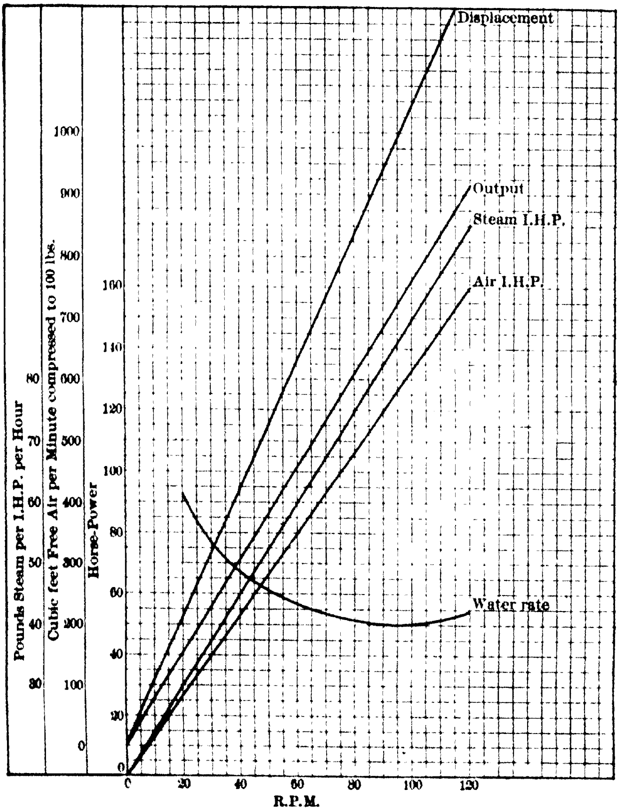


FIG. 90.—Compressor Plant No. 2.

platted, showing how the cost per steam horse-power per year is affected by the average running speed of the compressor. The curve of Fig. 89 shows the operating costs in another way. These costs may be read in terms of 1,000 cu. ft. of free air

compressed to 100 lbs. or 1,000 cu.ft. of compressed air at 100 lbs. gage pressure.

Test of Plant Number Two. The plant consisted of three 150 H.P. return tubular boilers, supplying steam for a Corliss

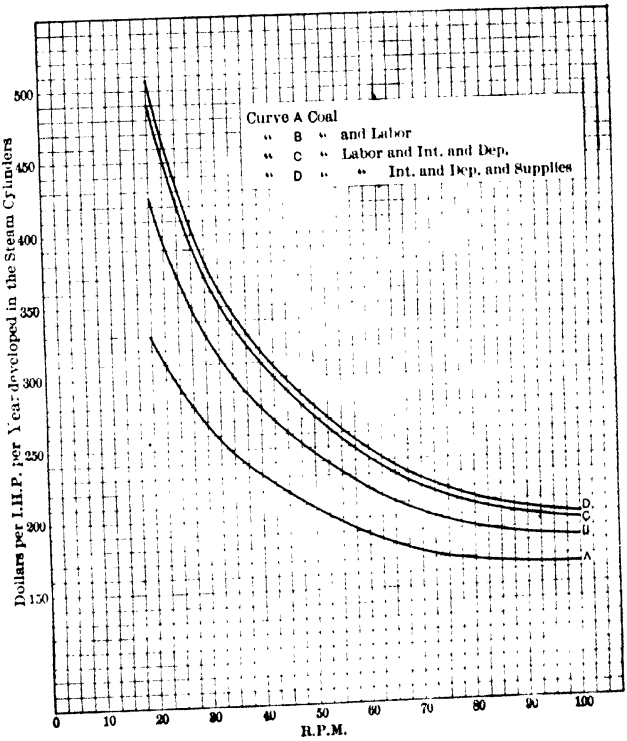


FIG. 91.—Compressor Plant No. 2.

engine, the air compressor, and steam heating. To determine the boiler horse-power, a meter was placed on the steam pipe to the compressor during the test run, so that only the portion of steam actually used by the compressor was charged to the same. The compressor was duplex, with Meyer valve

gear, simple steam cylinders 14 in. by 22 in., and two-stage air end, 14 in. and 22 in. by 22 in. stroke, rated by the manu-

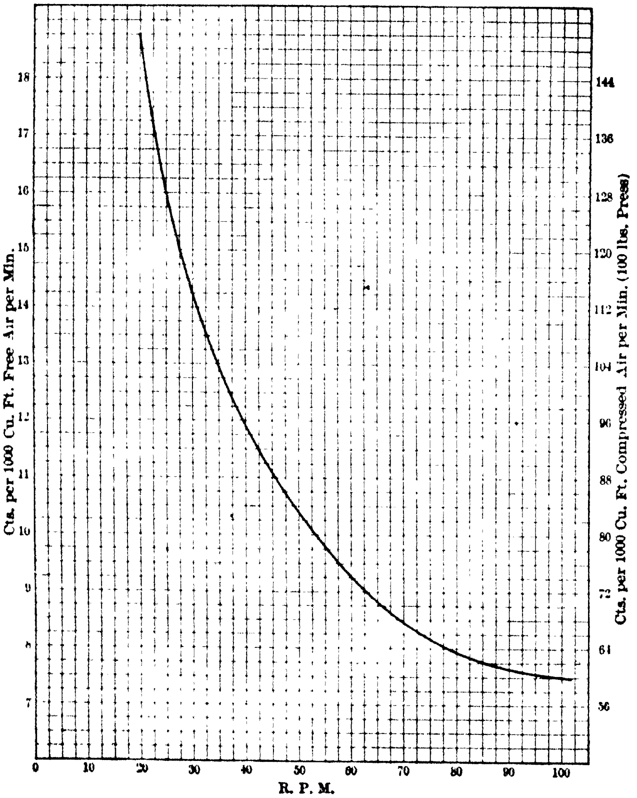


FIG. 92.—Compressor Plant No. 2.

facturer at 1,050 cu.ft. of free air per min. at 105 revs. At this plant the test lasted over a month, with the following results:

Total coal consumed, lbs.	459,250
Total feed water, lbs.	2,496,000
Average evaporation per lb. coal consumed, lbs.	5.46
Average revolutions per minute	36.
Corresponding average indicated horse-power (from curve)	53

Hourly readings of the revolution counter were taken, showing an average speed of 36.05 revs. At this speed the steam consumption was 51 lbs. per I.H.P. hour, as measured at the throttle, the air meter showing a delivery of 275 cu.ft. of free air per min. The total efficiency was 67%. Taking the ordinary method of computing the mechanical efficiency only at the same speed, there would be 48 air H.P., divided by 54 steam H.P., giving an efficiency of 89%.

The coal consumption per indicated horse-power per year, as shown by the books of the company, amounted at the average speed to about 56 tons. Table X, with Figs. 90, 91, and 92, present the details of the test on this plant, which was conducted in a manner similar to that on plant No. 1.

Test of Plant Number Three. This plant consisted of two 125 H.P. return tubular boilers, supplying steam for a noncondensing cross-compound air compressor of standard make; steam cylinders 18 in. and 35 in. by 24 in., air cylinders 14 in. and 28 in. by 24 in. A two weeks' run gave the following results:

Total coal burned, lbs	221,190
Total feed water, cu ft	34,273
Total feed water, lbs	2,094,657
Average temperature feed water, degrees F	154
Average evaporation per lb coal consumed, lbs	9 48
Average boiler horse-power	208
Average revolutions per minute	66
Average indicated horse-power of steam end, at 66 R P.M (from curve) . . .	210
Average indicated horse-power of air end (from curve)	128 5
Average steam pressure.	97
Average air pressure	97
Average outside temperature, degrees F	23
Average air piston displacement at normal speed, cu ft, at 70° F.	1,372
Metered output in cu ft corrected to 70° F	734

The average evaporation of 9.48 lbs. of water per lb. of coal, from 154° F. to an average steam pressure of 97 lbs., is equivalent to 10.4 lbs. of water evaporated from and at 212° F. At the average speed of 66 revs., the displacement was 1,240 cu.ft. of free air per min., while the metered output was 734 cu.ft. showing a net volumetric efficiency of 59%.

TABLE XI.—TEST ON PLANT NO. 3

COMPOUND NON-CONDENSING STEAM CYLINDERS, 18" AND 35" TWO-STAGE AIR END, 28" AND 14" BY 24" STROKE.

Revolutions per Minute.	Steam Pressure, Pounds.	Air Pressure, Pounds.	High-pressure Side. Steam M E P		Air. M E P		Low-Pressure Side Steam M E P.		Air M E P		Total Steam, I H P	Total Air, I H P	I H P Friction	Per Cent Loss in Friction.	Air Line Temperature.	Cu ft Displacement, 70° F	Output, Cu.ft Air per Minute 70° F.
			M E P	I.H P.	M E P	I H P.	M E P.	I H P	M E P	I H P.							
24	100	73	53.6	38.6	35.7	15.6	8.03	24.8	13.85	24.7	63.4	40.3	23.1	36.4	120	447	269
29.6	98	67	50.55	44.9	28.5	15.4	7.32	25.0	10.43	29.5	69.9	44.0	25.0	35.8	119	550	330
50	97	100	61.6	92.4	45.55	41.5	10.67	61.7	14.48	53.7	154.1	95.2	52.9	38.2	120	935	555
61	100	102	60.6	110.9	45.8	50.9	11.6	81.8	15.1	68.4	102.7	110.3	73.4	38.1	140	1145	677
75	99	100	62.6	140.0	44.68	61.0	11.76	102.0	15.36	85.5	242.0	146.5	96.4	39.7	160	1409	831
94.5	95	99	48.7	138.0	44.95	77.4	16.31	178.3	15.64	100.7	316.3	187.1	120.2	40.85	165	1780	1050
103.5	98	95	47.7	148	43.0	81.1	17.2	206	16.31	125.3	334.0	206.4	147.6	45.02	173	1048	1150

NOTE.—The above data are from a table of eighteen different readings taken at various speeds.

TABLE XII.—TEST ON PLANT NO. 4

DUPLEX COMPOUND NON-CONDENSING CORLISS CYLINDERS 18" AND 30", TWO-STAGE AIR END, 26" AND 16½" BY 30" STROKE

Revolutions per Minute	Steam Pressure, Pounds.	Air Pressure, Pounds	Temperature Air at Receiver	High-Pressure Side. Steam M E P.		Air. M.E.P		Low-Pressure Side Steam. M E P		Air M E P		Total Steam, I H P	Total Air, I H P	Loss in Friction, I H P	Friction Loss, Per Cent	Displacement, Cu.ft.	Output, Cu.ft Air per Minute, 70° F.
				M E P.	I.H P.	M.E.P	I H.P.	M E P	I H P	M E P	I.H.P						
20	153	60	98	44.7	34.08	29.55	18.3	8.46	18.08	14.82	23.6	52.16	41.9	10.26	19.67	365	342
33.3	155	61	75	48.73	61.8	31.28	32.4	6.50	23.1	14.85	39.4	84.9	71.8	13.1	15.44	612	570
50	152	71	78	53.1	101.1	38.98	60.5	8.03	42.9	15.02	59.9	144	120.4	23.6	16.37	922	850
66.7	149	85	88	59	149.8	43.73	90.6	8.54	60.9	13.15	70.10	210.7	160.7	50	23.71	1230	1118
75	140	78	94	54.15	154.7	37.52	87.4	7.93	63.5	16.64	90.5	218.21	186.9	31.3	14.34	1383	1249
90.9	145	57	128	49.8	172.3	27.52	77.75	0.76	94.8	10.14	138.6	267.1	216.35	50.75	19	1673	1490
100	140	73	109	53.2	202.6	26.15	81.2	11.07	118.4	10.16	152.8	321	234	87	22.1	1843	1625

NOTE.—The above data are from a total of thirty-six readings taken at various speeds.

To determine the conditions in average operation, the curve at 66 revs. should be followed (Fig. 93), at which the indicated horse-power of the steam cylinders was 210, and that of the air cylinders, 128. This shows the efficiency to be 61%, the friction loss being 81.5 H.P., or 39% of that delivered by steam end. This extremely high friction loss was due to the fact that the compressor shaft was out of line, and the plant could not be

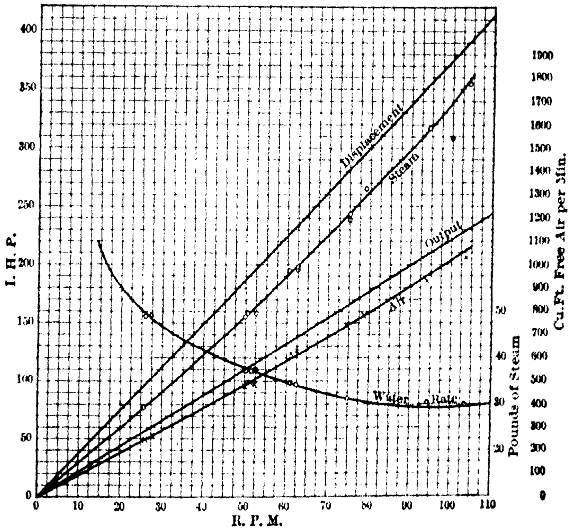


FIG. 93.—Compressor Plant No. 3.

shut down long enough to rectify it. The details and results of this test, given in Table XI and Figs. 93, 94 and 95, are interesting in exhibiting the inefficiency that may be caused by a purely mechanical defect.

Test Number Four. The results of a test on another plant are given in Table XII and Fig. 96, the details of the boiler test and of the costs being omitted. In this case the compressor was of the tandem compound non-condensing type, with Corliss valve gear for the steam cylinders. The test shows that, at a low

speed, the steam consumption increases more rapidly than with the Meyer type of valve.

Summary. The results of these tests are enlightening, in showing the actual amount of the losses occurring in the com-

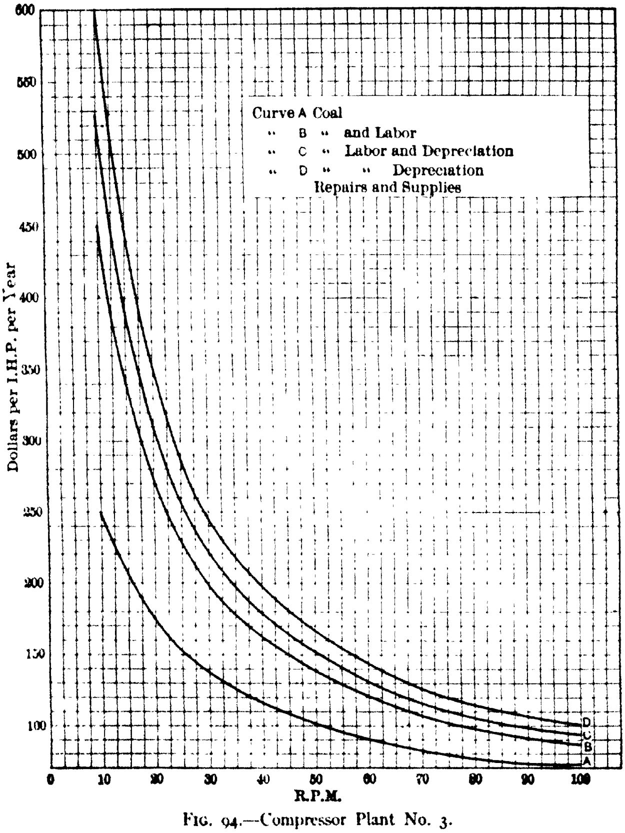


FIG. 94.—Compressor Plant No. 3.

pression of air, particularly when the compressor is operating under the unfavorable conditions of varying air consumption, unavoidable in mining and other work in which machine drills play an important part. These losses are always recognized as

existing, by compressor builders and by intelligent users, and it is clearly desirable that properly conducted tests should be made more frequently.

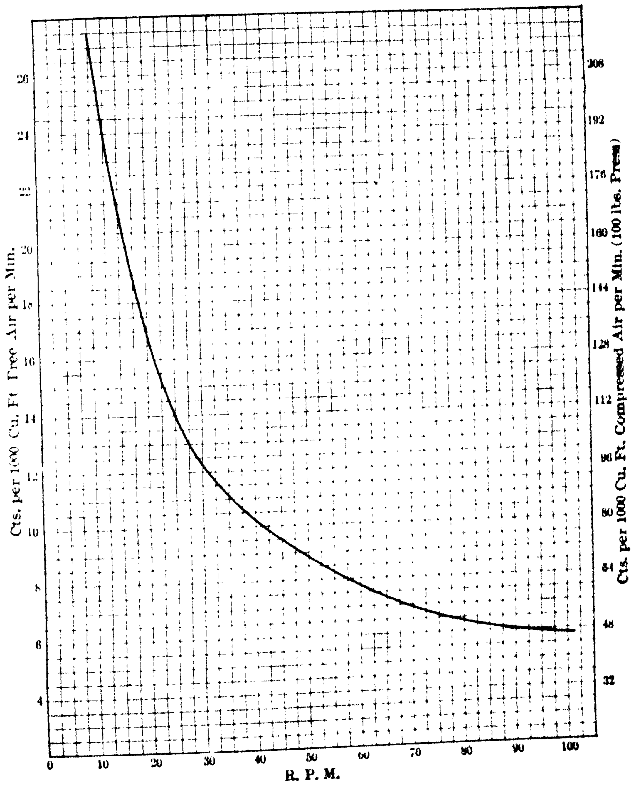


FIG. 95.—Compressor Plant No. 3.

Again, compressor plants generally develop less power than their full rated capacity. An air compressor is essentially a variable speed machine, its speed being regulated by some form of throttling governor, connected with the air-pressure regulator. It is therefore called on to run only as fast as the

demand for air may require. It would be well for compressor builders to give in their catalogues the horse-power rating at different speeds, with a table of efficiencies at different loads and speeds, just as is done by some of the manufacturers of

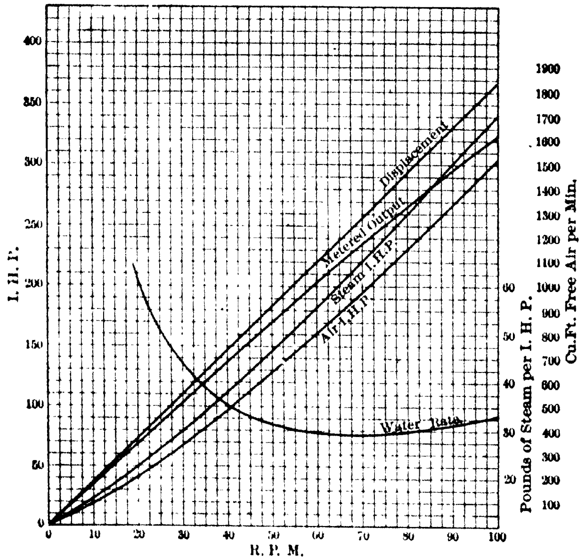


FIG. 96.—Compressor Plant No. 4.

electrical machinery. Catalogues might also include data respecting the cost per horse-power delivered by the air end of the compressor at different working speeds.

CHAPTER XI

AIR RECEIVERS

In its common form the receiver consists of a cylindrical shell of steel plate, resembling a steam boiler without tubes or flues. It has pipe connections to the compressor and air main, a pressure gage, safety-valve, drain cock, and man-hole. The vertical form, Fig. 97, is generally preferable, as it occupies less floor space. Fig. 98 shows a horizontal receiver. The capacity should be proportioned to the size of the compressor. The dimensions range from, say, 24 ins., diameter by 4 or 6 ft. long, to 48 or 60 ins. by 14, 16, or 18 ft., the largest sizes having a capacity of from 200 to nearly 400 cu.ft. Receivers are usually built to stand a test of 165 lbs. cold-water pressure, for working under pressure of 100–120 lbs., higher pressures than this being rarely necessary for mine service. The shells are single-riveted on circular seams and, except for small sizes, double-riveted on longitudinal seams; the heads being dished or hemispherical. For best results, the receiver should be placed close to the compressor, or in any case not more than 40–50 ft. distant.

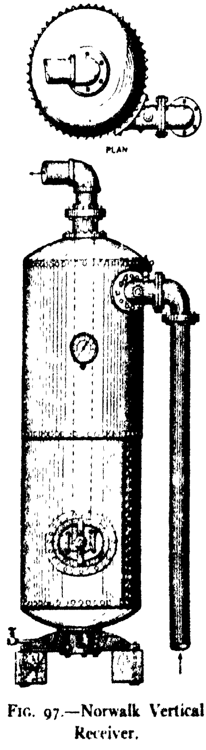


Fig. 97.—Norwalk Vertical Receiver.

The principal functions of an air-receiver are: (1) to eliminate the pulsating effect of the strokes of the compressor piston and

prevent rapid fluctuations of pressure; (2) to minimize the frictional loss attending the flow of air through the lines of piping; (3) to serve in some degree as an equalizer and reservoir of power; (4) to cool the air before it passes into the main, thus causing it to deposit a part of its moisture in the receiver, whence it is drained off.

Regarding the first point, the volume of the receiver should be sufficiently great in proportion to that of the compressor cylinder to prevent any material rise of receiver pressure by the volume of air forced into it at each stroke. If the compressed air passed directly into the main, large fluctuations of pressure would occur, accompanied by periodic acceleration of flow.

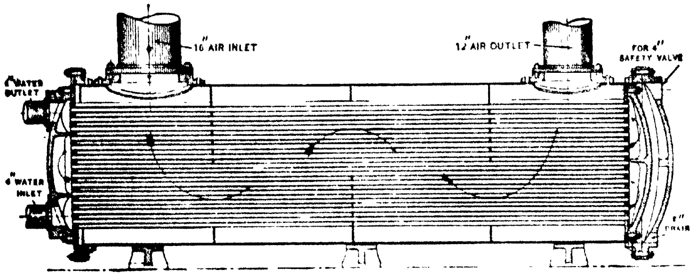


FIG. 98.—Horizontal Receiver-Aftercooler (Ingersoll Rand Co.)

This would increase the frictional resistance in the pipe, and at the end of each stroke the compressor piston would have to force the air out of the cylinder against a pressure momentarily greater than the normal. The violence of the discharge pulsations is obviously greater in a single cylinder than a stage compressor, because the total discharge must take place from a cylinder of larger diameter in a smaller proportion of the length of stroke than is the case with the high-pressure cylinder of a stage compressor. In the latter the delivery valves open earlier in the stroke, and the air pipe is about one-half the diameter of the cylinder.

The second function of the receiver is best fulfilled by placing an auxiliary receiver near the point where the compressed air is

used. Just as the receiver at the compressor diminishes the momentary rise of pressure in the main due to each stroke of the piston, so a second receiver close to the machine using the air prevents a drop of pressure as each cylinderful of air is drawn off. By reducing the fluctuations of pressure the two receivers maintain a practically constant flow of air through the main connecting them, thus minimizing friction and loss of pressure.* For mine service the second receiver would be placed somewhere underground; always an advantageous arrangement when the air main is long. Underground receivers are not often used for air drills alone, but they are a necessity for pumps and hoists run by compressed air. They are also useful in permitting a further deposition of moisture, thus rendering the air dryer and more suitable for expansive-working engines. To accomplish this most effectually, the underground receiver should be placed at the point in the pipe line where the air has reached its lowest temperature.

Underground receivers are usually made like those installed near the compressor. Sometimes, however, a chamber is excavated in the rock, and the walls cemented or asphalted tight. The chamber is closed by a brick dam of two parallel walls, with a 2-in. layer of cement between them. In the dam are set a cast-iron man-hole with suitable cover, the pipes for connecting with mains to the different working places, and a drain pipe and cock close to the floor. The latter is opened from time to time, to blow out the accumulated water and sediment. A pressure gage is attached to the man-hole cover. Such reservoirs may be built to cost much less (for large sizes) than ordinary shell receivers of equal capacity.

The third function of the receiver is apt to be exaggerated. While it acts to a limited extent as a reservoir of power; yet, to be of much practical use in this respect, it must be very large. For example, take a 20-in. compressor, working at 60 lbs. pressure. To meet the demand for only 1 minute after the compressor is stopped, and not have the pressure fall more than

* Questions relating to the flow of air in pipes, and frictional losses are discussed in Chap. XVI.

15 lbs., the receiver would have to be 5 ft. diameter by 50 ft. long. Again, if the compressor were running at a constant speed and the demand for air should suddenly increase 25%, as might happen in starting several more machine drills, a receiver of the size mentioned could meet the extra demand only 4 minutes. Long pipes of large diameter assist in equalizing the flow of air, but their use does not preclude the necessity of receivers. It is much cheaper to employ piping of moderate size, in connection with a receiver of generous dimensions.

The fourth function of the receiver is probably the most important. Considerable moisture is always present in compressed air, due to the natural humidity of the atmosphere, especially in warm weather. The velocity of the air coming from the compressor is greatly reduced on entering the receiver; and on cooling the air deposits part of its moisture, which otherwise would be conveyed into the piping, and thence to the machines using the air. Moisture tends to wash away the lubricant, and so increase wear, and consequent leakage of air and loss of economy. This is especially true of high-speed machines, as drills and small air hoists, in which the wearing surfaces are limited in area. Moisture collecting in pipe lines also causes " water hammer," reduces the air passage by accumulating at low points, and in winter may freeze and burst the pipes. Wet air, freezing in drills, etc., may clog the exhaust and increase back pressure. Moreover, hot air in pipe lines causes expansion, and when the pipe cools during a shut-down, contraction takes place, all of which tends to leaky joints. The receiver should be large enough to drain the air thoroughly. In the ordinary sizes of receiver the results are usually quite imperfect, because the air passes too rapidly to permit a large drop in temperature. The inlet and outlet pipes of the receiver should be placed in proper relative positions. If at opposite ends, and especially if the pipes point toward each other, a strong through current is caused, and the air passes out without having had time to cool or to drop much of its entrained moisture. One mode of arranging the pipe connections is to place the inlet on one side, near the end of the receiver, while the outlet is at the opposite

end, in the middle of the head. The air is thus forced to change its direction of flow. Or, as in Fig. 97, both pipes may be connected near the top, the outlet pipe being carried nearly to the bottom, where the air is likely to be slightly cooler (and dryer) than at the top. As the inlet pipe shown in this case is connected tangentially to the periphery of the receiver, a rotary motion is imparted to the body of air, so that each particle remains longer in the receiver and under its cooling influence. Some receivers have baffle-plates for the same purpose, as in Fig. 98. Part of the lubricating oil carried over from the compressor cylinder is also deposited in the receiver. At intervals, according to atmospheric and other conditions, the water and oil are drained off.

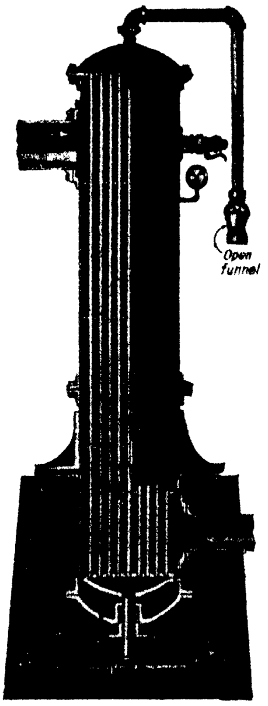


Fig. 99.—Ingersoll Rand Vertical Aftercooler, Type "VK."

Receiver Aftercoolers. A receiver of ample size, placed close to the compressor, tends to economize power; because, whatever cooling is accomplished reduces the temporary increase of pressure due to the heat of compression. Hence, in forcing the air out of the cylinder against the receiver pressure, the piston consumes less power than if the air were left to cool gradually in a long length of piping. As the heat of compression is always lost before the air is used, this saving is worth while, however small it may be, since it is produced without cost. This consideration has of late led to the employment of "receiver aftercoolers," Fig. 98 shows one form, somewhat similar to the

intercooler in Fig. 49.* Fig. 99 shows a recent design, made in 6 sizes, from 20½ ins. diameter by 10⅛ ft. long, to 45¼ ins. by 19 ft.; cooling surface, 152–2012 sq. ft. Horizontal coolers are also furnished.

These aftercoolers cool and dry the air more thoroughly than ordinary non-tubular receivers, and so minimize the troubles referred to on p. 172. In Fig. 99 the shell is steel, with cast heads. The arrows show the direction of flow of air and water. By means of the open funnel in the water discharge pipe, the flow of water is seen, and regulated as necessary. A plate in front of the air discharge prevents escape of water deposited by the air. Such an aftercooler will reduce the temperature of compressed air to within 15° or 20° of that of the entering water. The Ingersoll-Rand Co. gives the following figures for air at 80–100 lbs., from a two-stage compressor:

Temp. Cooling Water	Gals. per hr. per 100 cu. ft. free air per min.
50° F	120
60° F	150
70° F	180
80° F	210

Capacity of Receivers. No exact rule can be given. Referring to the statements on pp. 171 172, regarding the third function of receivers, the capacity should be sufficient to prevent rapid or great fluctuations of air pressure, and must therefore depend largely on the kind of service and local conditions.

A safe rule for ordinary rock-drill service is to allow a receiver capacity of 100 cu.ft. per 800 to 1000 cu.ft. of free air compressed per minute For stationary, constant-running engines, like pumps, the capacity may be smaller.

* See an article by Frank Richards, in *Compressed Air*, Jan., 1907, p. 4329.

CHAPTER XII

SPEED AND PRESSURE REGULATORS FOR COMPRESSORS

If the air consumption were constant, no more regulation of the compressor's speed and power would be required than that furnished by the steam governor, to take care of fluctuations in boiler pressure or accident to the mechanism. But there are usually wide variations in the rate at which the air is used. In event of a sudden decrease in consumption, the compressor must be slowed down, or air will be blown off at the receiver safety valve. Since a cubic foot of compressed air costs more than a cubic foot of steam, the compressor must have some device for coordinating the quantity of steam admitted to the steam end with the variable receiver pressure, thereby regulating the piston speed in accordance with the demands upon the air end. Furthermore, it is not enough to provide only for varying the speed of the compressor. At times, the consumption of air may cease entirely for a short period, and, to avoid bringing the compressor to a standstill, provision should be made for unloading the air end. Useful work then stops for the time being, the compressor consuming only enough steam to turn its centers.

Numerous regulating and unloading mechanisms have been devised, so that instead of requiring the constant attendance of an engineer, the compressor operates automatically under wide variations of load. These devices may be classified under two heads: (1) speed governors and pressure regulators; (2) unloaders for the air cylinders.

Speed Governors and Pressure Regulators. Speed governors, usually of the centrifugal or flyball type, may be applied to the steam cylinder merely to regulate speed; or their action may be controlled by the receiver pressure so as to regulate both speed and pressure. The air cylinder is not completely

unloaded at any time, the compressor being simply speeded up or slowed down according to the rate at which the air is used.

The flyball governor of the regulator type is illustrated by Fig. 100. The stem *h* of the throttle valve connects with the spindle of the ball governor, by which the speed of the compressor is controlled. At *p* is the bevel gearing for driving the governor, a small pulley being mounted on the gear shaft and

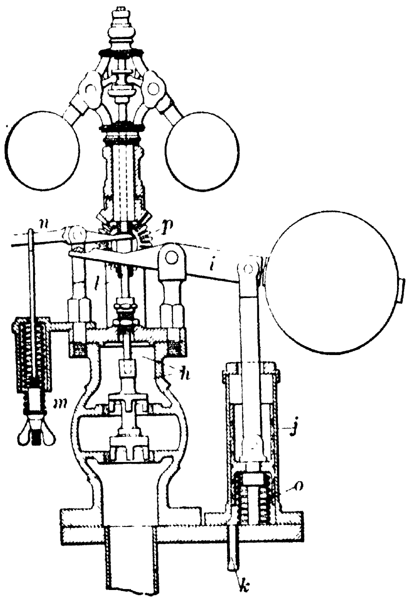


FIG. 100.—Clayton Governor and Pressure Regulator.

belted to the crank-shaft of the compressor. The action of the ball governor is modified by the weighted lever *i* and the air cylinder *j*, which is connected to the air receiver by the pipe *k*. When the receiver pressure exceeds its assigned limit, it raises the piston and weight, and shuts off steam by forcing down the throttle valve *h*, the pressure of the lever being applied at *l*. The governor is adjusted to its work by the spring and thumbscrew *m*, acting on the lever *n*, which tends to keep open the

throttle against the downward pressure of the lever *i* upon the valve stem. The spring *o* eases the drop of the weight when the air pressure falls.

Flyball governors of several types are used on the Ingersoll-Rand, Sullivan, American and other compressors.

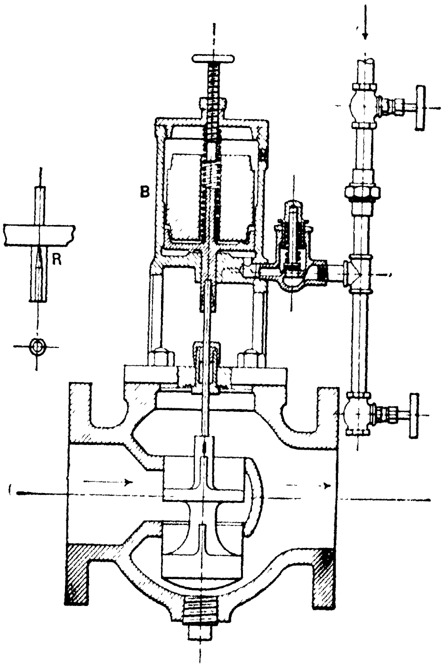


FIG. 101 —Norwalk Pressure Regulator.

The design of the Norwalk governor (Fig. 101) is entirely different. Above a balanced throttle valve in the main steam pipe is set a small air cylinder B. At the side of the cylinder is a spring-controlled valve, connected by a pipe with the receiver, or with the air main leading to it. The spring of this valve is adjusted so that the air will lift the valve, and pass through it, at any desired pressure. When the receiver pressure exceeds

this limit the valve allows air to pass under the heavy piston in the cylinder B, raising it and partly closing the throttle. If no escape were provided the piston would be forced at once to the top of the cylinder. To regulate its movement and prevent shutting off the steam completely, a tapered recess is cut in the piston rod of this cylinder, at the point where it passes through the lower-head (indicated at R, in the small cut to left of main figure). As the piston is forced upward by the air pressure the area of the opening formed by the slotted stem furnishes a graduated escape for the air, and so regulates the small piston's movement and through it the position of the throttle valve. The upward movement of the piston is still further regulated by the screw stop and spring in the top of the cylinder. This can be so adjusted that, when the piston reaches its highest point, the throttle valve still admits enough steam to keep the compressor turning its centers.

With governors of the preceding types, the operation and control of the compressor is not automatic under all conditions, but they answer the purpose for some kinds of service. In case no air is drawn from the receiver, the compressor is brought nearly to a standstill; then, if the pressure continues to rise, a little air will blow off at the receiver safety-valve, or the compressor may be stopped by closing the throttle.

A combined governor and pressure regulator, with unloading attachment, as employed by the Sullivan Machinery Co. for steam-driven compressors, illustrates a mode of control that has been adopted by several builders, though with many variations in details (Fig. 102). The split-ball governor (11), belt-driven from the crank-shaft to the pulley (29), accompanied by the tightener (43), controls the steam throttle (3). Connected with the governor spindle and throttle valve stem, at (28), is a lever (25), the position of which is influenced by the centripetal action of the set of springs (31, 32, and 26). By screwing up or down the hand-wheel and speeder screw (5), this system of springs (and with them the governor) is set to run the compressor at any desired speed. The other element of the governor is the air-pressure device, which, by the position of the plunger in the

small air cylinder (18), brings the springs into action in the order of their strength, thus producing movement of the lever (25). The pressure device is connected to the receiver by the union valve (33), admitting air to the little cylinder (27), the piston of which operates a needle valve. This valve is held closed against any desired minimum air pressure by the adjustable weight (36) and the regulating screw and spring (37 and 38).

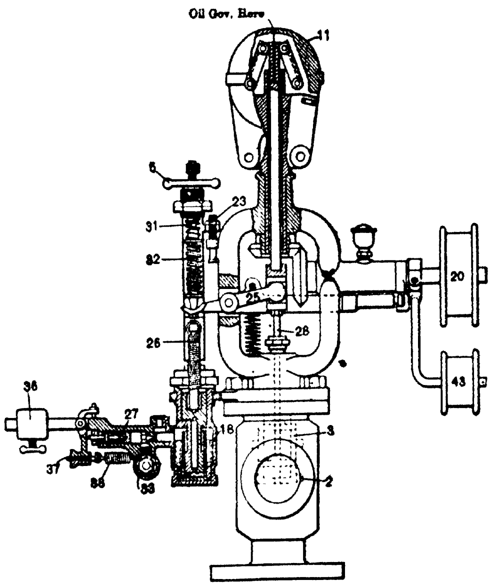


FIG. 102.—Sullivan Governor and Unloader.

When the receiver pressure exceeds the normal, it opens the needle valve and admits receiver air to the cylinder (18). As the pressure increases, the plunger in (18) rises against the counter-spring (26) and through the lever (25) tends to close the main steam throttle (3), thus slowing the compressor. Total stoppage is prevented by screwing down the nut of the stop-screw (23), so as to limit the upward movement of the plunger (18), which acts intensively, being so proportioned that a variation of only 2 or 3

lbs. receiver pressure is multiplied to say 40 lbs. in its action on the governor. A sensitive control is thus produced within narrow limits of working pressure. To prevent violent movements of the pressure element, in case of sudden changes of receiver pressure, the plunger in (18) has an oil dash-pot.

A somewhat similar pressure regulator and unloader is used on the Franklin compressor.*

For steam-driven compressors of the Corliss type, as built by the Ingersoll-Rand, Nordberg, Laidlaw-Dunn-Gordon, Sullivan, Allis-Chalmers, and some other companies, the regulators act in conjunction with ball or other centrifugal governors. They control by changing the point of cutoff in the steam cylinder.

The Laidlaw-Dunn-Gordon governor (Fig. 103) is an example. Air from the receiver enters the small cylinder A, the piston of which is weighted. The action of the lever B is adjusted by the coil spring C. This lever is linked to a floating lever D, pinned to the vertical side rods of the ball governor. D is connected by the link E to the bell-crank F, the lower arm of which is connected through the long horizontal rod G to the Corliss steam gear. By this system of levers, the movement of G, and through it the point of cutoff, is under the combined control of both ball governor and of the receiver pressure, as influencing the position of the piston of the cylinder A. The arm H is pivoted at the foot of the governor post. Connected to it are the cam I and the idler pulley J, which rests on the governor belt. In case the belt breaks, the idler pulley falls and the cam allows the governor to drop, thus shutting off steam and preventing the compressor from racing.

One of the Ingersoll-Rand regulators, used for compressors with compound steam ends, controls speed by varying the cutoff of the high-pressure cylinder (Figs. 104, 105). This governor contains an oil pump *a*, chain-driven from the compressor crank shaft to sprocket *s*. The oil from the pump enters the plunger chamber *c* under pressure, and acts to force upwards the plunger *d*, which carries weight *e*. Vertical movement of *d* is transmitted by rack *g*, pinion *h*, and sprocket *i*, through a

* *Mines and Minerals*, May, 1905, p. 504.

chain, to the cutoff valve stem, causing the stem to rotate and thus vary the cutoff. Admission of oil to *c* is controlled by a by-pass valve *f*. This valve is operated through the panto-

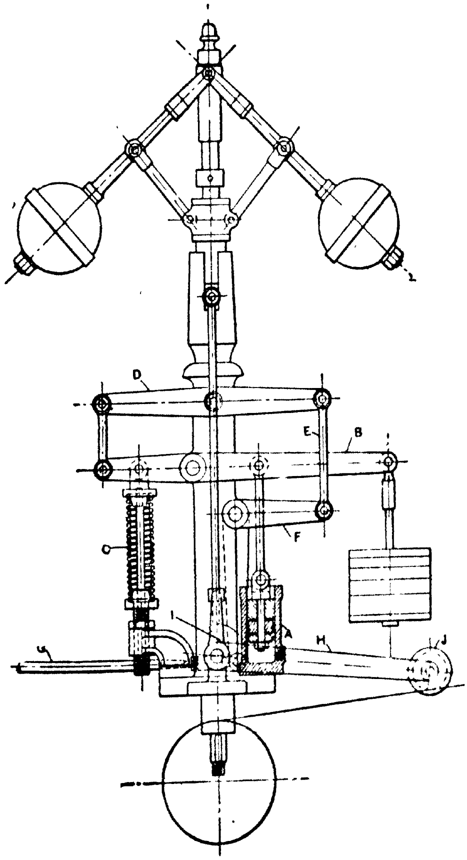


FIG. 103.—Laidlaw-Dunn-Gordon Air Governor.

graph motion *j*, by the movement of either the plunger *d*, or the air regulator weight and lever *k*.

When the compressor speed (and therefore the speed of the

oil pump) produces sufficient oil pressure to force plunger *d* upward, valve *f* opens and checks the rise of *d* until the speed increases still further. The maximum and minimum speed screws *l* and *m* are set at the factory to limit the maximum running speed and to prevent such a shortening of the cutoff as would stop the compressor.

Air-Cylinder Unloaders. These exercise complete control when the compressor is belt-driven, and also for steam-driven

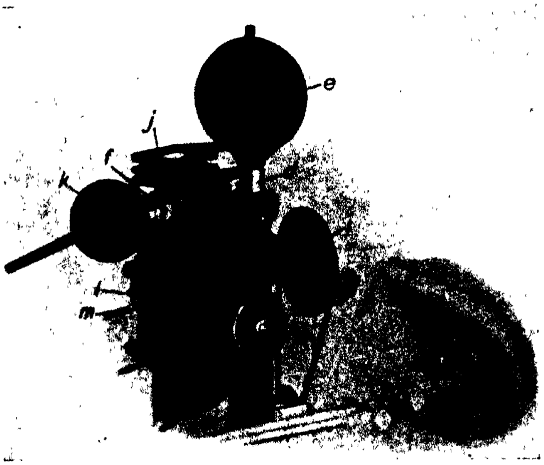


FIG. 104.—Ingersoll-Rand "XPV" Automatic Air Governor

compressors when used in conjunction with a governor. In steam compressors, as the consumption of air decreases the throttle is first nearly closed; then, if it ceases altogether, the unloading mechanism either shuts off the intake air or holds open the discharge valves, thus admitting air at receiver pressure to both ends of the cylinder. In either case the pressures on opposite sides of the piston are balanced and useful work ceases, though the compressor continues to run slowly.

The Rand "Imperial" unloader, for compressors driven by a belt or direct-connected electric motor, is an example of this

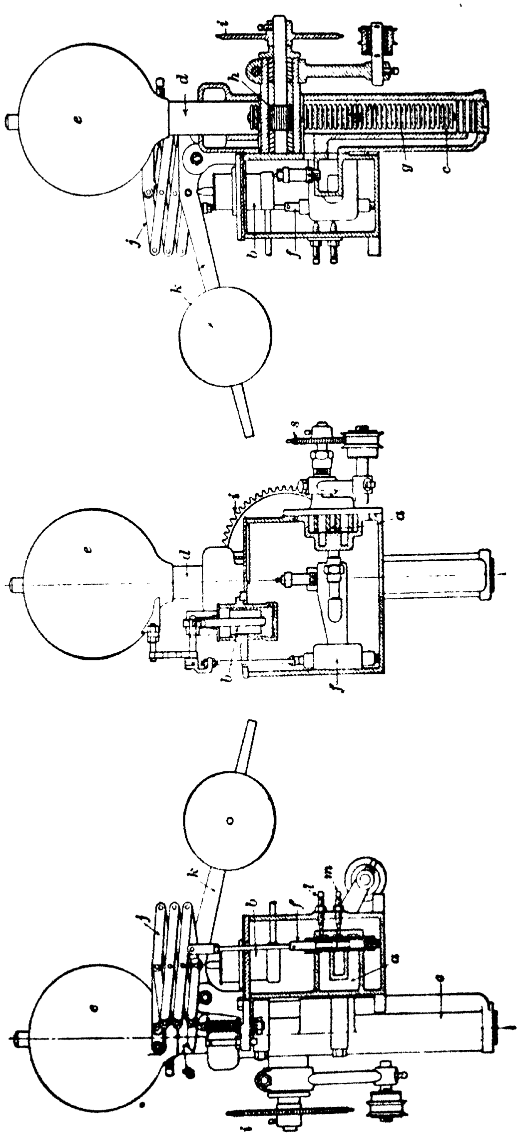


Fig. 105.—Details of Ingersoll-Rand "XPV" Automatic Air Governor

type of regulator. It is placed in the intake pipe, and shuts off the air from the inlet valves when the receiver pressure rises above the set limit. In Fig. 106 the the intake air enters as shown by the arrows. The small chamber (60) is connected

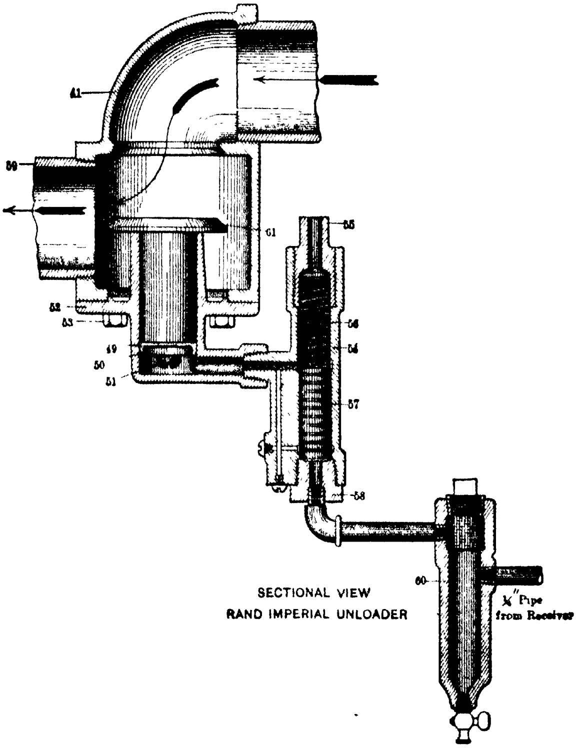


FIG. 106.

by a ¼-in. pipe with the receiver. As the pressure increases, the piston (57) moves against the spring (56), admitting receiver air through the small ports on the left of the piston to the lower side of the plunger valve (61). On reaching its seat this plunger closes the air intake. The spring (56) may be adjusted by the

screw-plug (55) for any required working pressure. As the receiver pressure falls, on increased consumption of air, the spring forces down the piston (57). This closes the lower small air port, leading to the under side of the plunger valve (61), and opens the upper horizontal port, connecting with the open screw-plug (55). The air below the plunger valve is thus exhausted, causing the latter to reopen the intake passage. The compressor then resumes useful work. An unloader similar to the above is used in some of the Allis-Chalmers compressors. An automatic " choking " controller is shown in Fig. 29, Chap. II. The Ingersoll-Rand Co. also makes a clearance controller, for unloading the air end of small power-driven compressors It varies the clearance volume of the cylinder by cutting in or out some of the discharge valves. A small air cylinder, connected with the receiver, is attached to the compressing cylinder. As the receiver pressure increases, the weighted piston of the controller cylinder rises higher. Inserted in the side of this cylinder is a series of small pipes, each connected by branches with a discharge valve on each end of the main cylinder. These valves are thus released from the receiver pressure successively, as the pressure increases, and the work done by the compressor is proportionately reduced. When normal receiver pressure is restored, the valves close automatically, and compression and delivery are resumed. This controller has the disadvantage of suddenly releasing and resuming the load.

Another type of unloader is employed on the Nordberg constant-speed, variable-delivery compressor. It is for motor-driven machines, with Corliss air valves, and operates by closing the inlet valve before the forward stroke is completed. During the remainder of the stroke, the air already admitted to the cylinder expands below atmospheric pressure, and is then compressed on the return stroke. This is practically equivalent to varying the working length of stroke.

The valve gear of this compressor is shown in Figs. 107 and 108. In Fig. 107 the wrist-plate *w* is driven by the rod *a* from an eccentric on the fly-wheel shaft; another eccentric operates the releasing mechanism through the rod *b*, which oscillates

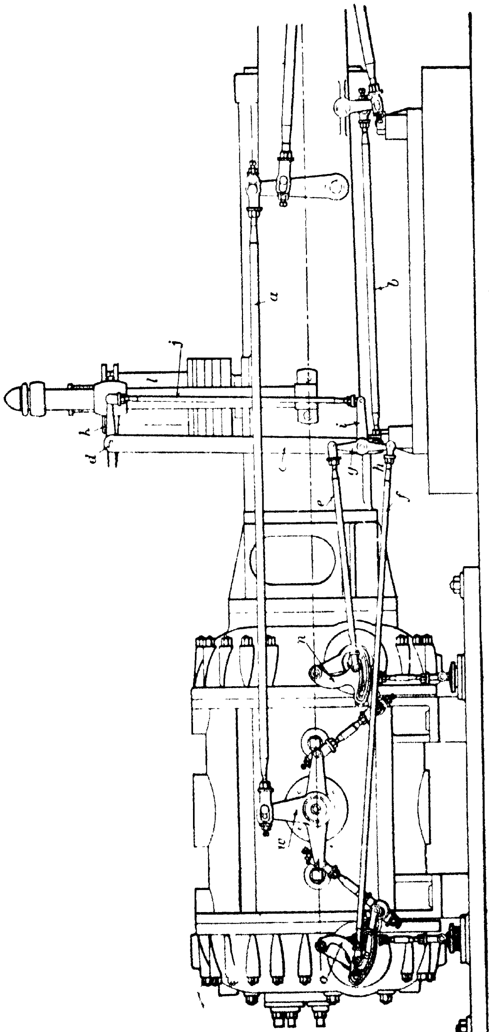


Fig. 107.—Valve Gear of Nordberg Constant-Speed, Variable-Delivery Air Com-pressor.

the arm c about the fixed center d. Swivelled to the lower end of c is a 3-armed rocker. The arm i is linked by the rod j to the radius fork k, which in turn is connected to the pressure governor l. The arms g and h of the rocker, through the rods e

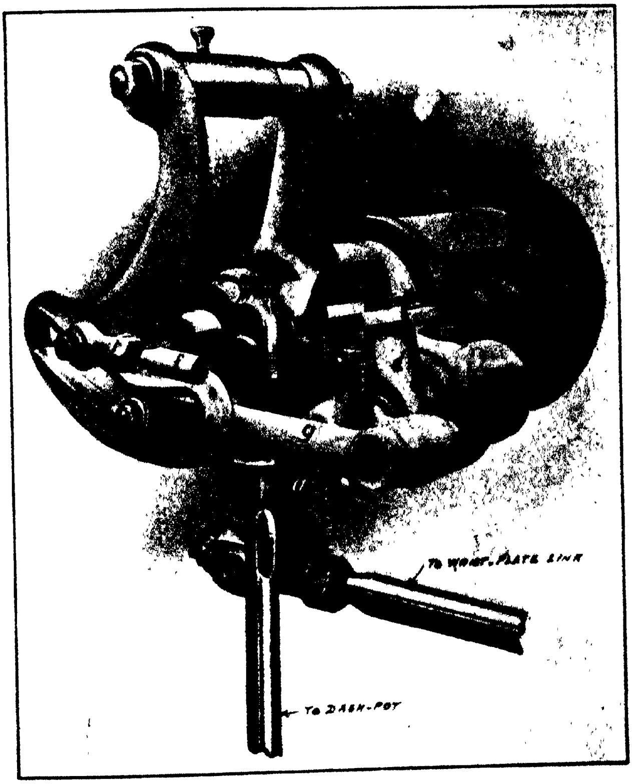


FIG. 108.—Detail of Valve Gear Shown in Fig. 107.

and f, operate the knock-off or releasing cams n and o, attached to the inlet-valve spindles. When the compressor is working regularly, under normal air consumption, the rocker arms g and h remain vertical, under the action of the eccentric rod b, and

impart equal movement to both knock-off cams. If, however, the receiver pressure increases, the rocker arm *i* moves upward, the arms *g* and *h* take an inclined position and, through the rods *e* and *f*, the point of release of the valves is altered.

The releasing mechanism is shown by Fig. 108. Mounted on the valve spindle is a rocker having three arms, *a*, *b*, and *c*. The wrist-plate link is connected to arm *a*, the releasing latch *d* to arm *b*, and the governor cam-arm *e* to arm *c*. By the rod *f*, *e* is connected also to the governor as explained above, and hence has a compound motion; it swings bodily above its swivel pin at the top, and its position is adjusted laterally by the action of the governor. The cam slot has two circular arcs, struck from the center at the upper end of *e*, with an inclined jog connecting them. Since the roller on the arm *g* swings about its center under the action of the cam groove, as the cam is moved from the main eccentric by the rod *f*, the latch *d* is alternately released and engaged, when the roller passes the jog in the cam. The point of the stroke at which release takes place is determined by the governor.

Figs. 109, 110 and 111, are a set of indicator cards from a two-stage compressor provided with this regulating mechanism, and running at 74 revs. per min. The upper card in each cut is from the intake cylinder, the lower from the high-pressure cylinder. Fig. 109 shows the cards when working at nearly full load. Fig. 110 (half load) illustrates the action of the regulating gear. Taking the crank-end card C, the inlet valve remains open from the beginning of the stroke, at *a*, approximately to mid-stroke *b*, at which point the releasing gear acts and the valve closes. From *b* to the end of the stroke, at *c*, the air in the cylinder expands below atmospheric pressure. On the return stroke, the compression line nearly coincides with the expansion line from *c*, until atmospheric pressure is reached at the point *b*, after which compression proceeds as usual. The action of the inlet valves of the high-pressure cylinder is the same, except that the expansion and re-compression of the air is from receiver pressure, instead of atmospheric pressure. In Fig. 111 the cards show the small amount of work done when the compressor

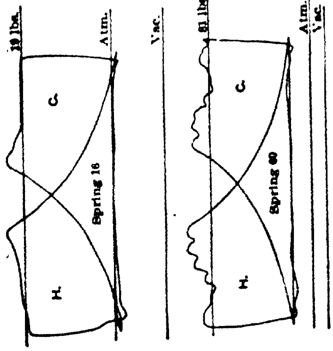


FIG. 109.—Nearly Full Load.

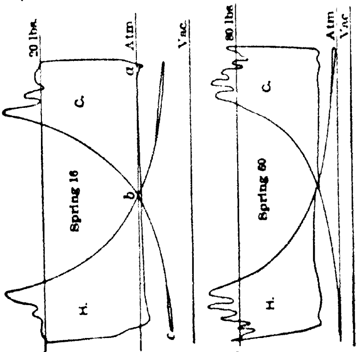


FIG. 110.—Half Load.

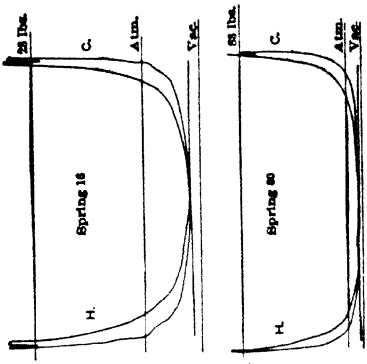


FIG. 111.—Nearly Zero Load.

is under nearly zero load. To simplify the mechanism, each cylinder has its own governor.

The Ingersoll-Rand "RA-39" controller (Fig. 112) is another device for cutting off air at the intake. It consists of a balanced disk valve *a*, inserted in the intake pipe and held open by the spring *g*. When the receiver air, entering at *e*, exceeds the desired pressure, it forces the diaphragm *c* to the right against the spring *f*, the resistance of which is adjusted by the screw behind it. Attached to the diaphragm is a needle valve *b*, which admits receiver air against the hollow-piston *d*,

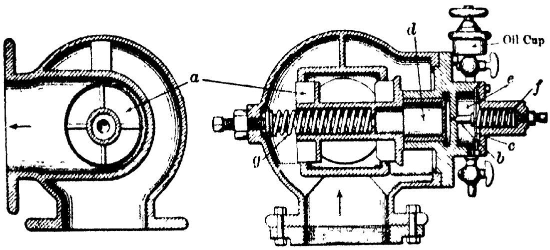


FIG. 112. Ingersoll-Rand "RA-39" Controller

thus closing the valve *a*. When the receiver pressure falls, the needle valve closes, the air leaks out from behind the piston *d*, and the regulator valve is forced open by spring *g*, again admitting air to the compressor. This controller is applicable to belt- or motor-driven compressors. It may be used in connection with the fly-wheel governor for steam-driven compressors. When a variable speed is desired, a fixed cutoff with a variable speed throttling governor is substituted.

CHAPTER XIII

AIR COMPRESSION AT ALTITUDES ABOVE SEA-LEVEL

BECAUSE of the diminished density of the atmosphere, air compressors do not produce the same results at high altitudes as at sea-level. Their effective capacity is reduced because a smaller weight of air is taken into the cylinder at each stroke. It is necessary, therefore, to modify the figures relating to the capacity and performance of compressors, as set forth in the first part of Chap. X. This matter is of especial importance in connection with mining operations, because of the large number of mines situated in elevated mountain regions. The rated capacities of compressors, in cubic feet of air, as given in the makers' catalogues, are for work at normal atmospheric pressure, and due allowance must be made for decreased output at elevations above sea-level. This reduction in output, which is usually also tabulated in handbooks and catalogues, should receive due consideration in order to avoid serious errors. For example, the volume of compressed air delivered at 60 lbs. pressure, at 10,000 ft. elevation, is only 72.7% of the volume delivered at the same pressure by the same compressor, at sea-level. In other words, a compressor which at sea-level will supply power for 10 rock-drills, will at an elevation of 10,000 ft. furnish air for only 7 drills.

The foregoing statement relates only to the volumetric capacity of the compressor. It must be remembered that the heat of compression increases with the ratio of the final absolute pressure to the initial absolute pressure. As this ratio increases with the altitude, more heat will be generated by compression to a given pressure at high altitudes than at sea-level. This additional heat temporarily increases the pressure of the air

in the cylinder, while under compression, and more power is therefore required to compress and deliver a given quantity of air. The corresponding loss of work, due to the subsequent cooling of the air in receiver and piping, also increases with the altitude.

Contrary to a common impression, the volume of air delivered by a given compressor does not bear a constant ratio to the barometric pressure, but at different altitudes this volume decreases slower than the barometric pressure. This relation may be shown as follows:* Two ideal indicator cards are

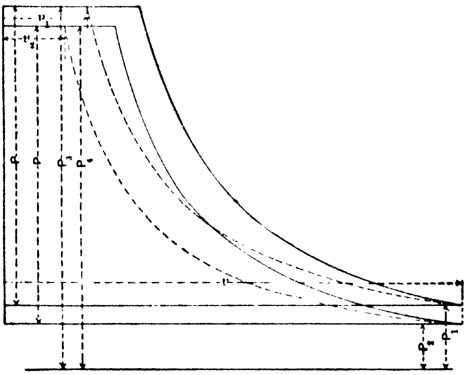


FIG. 113

represented in Fig. 113, one of a compressor working at sea-level, with an initial pressure P_1, the other at an altitude with a lower initial pressure P_2. The initial volume V and the final gage pressure P are the same for both compressors, P_3 and P_4 being the respective final absolute pressures. V_1 and V_2 are the final volumes, corresponding to the dotted isothermal curves, these volumes being taken as the basis, because they are those

* The general method of demonstration here given, together with Fig. 113 and accompanying table, are taken by permission from an article by F. A. Halsey, in *American Machinist*, June 2, 1898, p. 27.

to which the compressed air will eventually shrink on losing the heat of compression. From the theory of air compression,

$$VP_1 = V_1P_3, \text{ or } \frac{V}{V_1} = \frac{P_3}{P_1}, \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and

$$VP_2 = V_2P_4, \text{ or } \frac{V}{V_2} = \frac{P_4}{P_2}. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

But since $P_3 = P_1 + P$, and $P_4 = P_2 + P$, equations (1) and (2) may be written:

$$\frac{V}{V_1} = \frac{P_1 + P}{P_1} = 1 + \frac{P}{P_1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and

$$\frac{V}{V_2} = \frac{P_2 + P}{P_2} = 1 + \frac{P}{P_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Dividing equation (3) by equation (4):

$$\frac{V_2}{V_1} = \frac{1 + \frac{P}{P_1}}{1 + \frac{P}{P_2}}, \text{ or } V_1 : V_2 :: 1 + \frac{P}{P_2} : 1 + \frac{P}{P_1}. \quad . \quad . \quad (5)$$

This gives an expression for the ratio between pressure and volume at sea-level and for any altitude above sea-level, of which the corresponding barometric pressure is P_2. Thus, let $P_2 =$ 10 lbs., $P = 90$ lbs., and V_1 (from Table VII, p. 145) $= 0.1404$ cu.ft. By substituting these quantities in equation (5), V_2 is found to be 0.0999, or approximately 0.1 cu.ft.

In Table XIII, columns 4 and 5, are given the relative volumetric outputs, at gage pressures of 70 and 90 lbs., of a compressor working at different altitudes, the figures being percentages of the normal output at sea-level. These percentages have been derived by Mr. Halsey from equation (5), a constant loss of initial pressure of 0.75 lb. being assumed, to allow for the resistance presented by the inlet valves (see Chap. VII); that is, for practical purposes the sea-level atmospheric pressure is taken as 14, instead of 14.7 lbs. The figures in columns 4 and 5, which are for the ordinary range of pressure

employed in mining, show that, though there is a difference of 20 lbs. between the two gage pressures, the outputs vary only by a few thousandths and may often be neglected.* Wide differences, however, occur in the other columns. The method of computing compressor horse-power for a given number of machine drills, working at altitudes above sea-level, is given in Chap. XX, p. 298.

TABLE XIII

Altitude Ft.	Barometric Pressure. Inches Mercury	Barometric Pressure. Lbs per Sq in	Relative Output for Gage Pressure 70 lbs	Relative Output for Gage Pressure 90 lbs	M. E. P. for Gage Pressure 70 lbs	M. E. P. for Gage Pressure 90 lbs	Cu ft Piston Displacement per I H P for Gage Pressure 70 lbs	Cu ft Piston Displacement per I H P for Gage Pressure 90 lbs	Cu ft. Compressed Air per I H P for Gage Pressure 70 lbs	Cu ft. Compressed Air per I H P for Gage Pressure 90 lbs
1	2	3	4	5	6	7	8	9	10	11
0	30.00	14.75	1.000	1.000	33.1	38.2	6.93	5.99	1.144	801
1,000	28.88	14.20	967	966	32.6	37.6	7.03	6.09	1.123	787
2,000	27.80	13.67	935	933	31.1	36.9	7.15	6.20	1.103	773
3,000	26.76	13.16	904	900	31.5	36.3	7.27	6.31	1.084	759
4,000	25.76	12.67	873	869	31.0	35.6	7.39	6.43	1.065	746
5,000	24.70	12.20	843	839	30.5	35.0	7.51	6.55	1.046	733
6,000	23.86	11.73	813	809	30.0	34.3	7.65	6.67	1.028	720
7,000	22.97	11.30	785	780	29.4	33.7	7.80	6.79	1.011	708
8,000	22.11	10.87	758	751	28.9	33.1	7.94	6.92	994	695
9,000	21.29	10.46	731	723	28.3	32.5	8.09	7.06	976	683
10,000	20.49	10.07	705	696	27.8	31.8	8.24	7.20	959	670
11,000	19.72	9.70	680	671	27.4	31.2	8.39	7.34	942	658
12,000	18.98	9.33	656	647	26.9	30.6	8.54	7.49	925	646
13,000	18.27	8.98	632	623	26.3	30.0	8.71	7.64	908	635
14,000	17.59	8.65	608	600	25.8	29.4	8.88	7.80	891	624
15,000	16.93	8.32	585	576	25.3	28.8	9.06	7.96	875	613

Owing to the increase of piston displacement per indicated horse-power, as shown in columns 8 and 9 of the table, some builders make the air cylinders of compressors for mountain work of larger diameter for the same size of steam cylinder than those for sea-level service. As against the losses of the air end of the compressor at high altitudes, there is some gain in mean-effective pressure of the steam cylinders, because the exhaust

* For this reason, in compressor-builder's catalogues, no account is taken of the gage pressures in tables of compressor capacities at altitudes.

takes place against a lower atmospheric pressure. The same is true in part of the exhaust of machines using the compressed air. But the resultant of these gains is small and cannot be given much weight in offsetting the losses. A large deduction, for example, would have to be made for the lower calorific power of a given fuel at high altitudes.

The relation between compressor output and barometric pressure may be expressed simply in another way. Take the case of two compressors of the same size, one operating under an atmospheric pressure of, say, 14 lbs. and the other at 10 lbs. (corresponding approximately to an altitude of 10,000 ft.) If the first compressor is producing 6 compressions, the final absolute pressure will be $14 \times 6 = 84$ lbs. or about 70 lbs. gage pressure. To produce the same gage pressure, the other compressor must work to an absolute pressure of $70 + 10 = 80$ lbs., the number of compressions corresponding to which is $\frac{80}{10} = 8$. From each cubic foot of free air the first compressor will produce $\frac{1}{6}$ of a cu.ft. of compressed air, and the second compressor, $\frac{1}{8}$ cu.ft. Hence, the ratio of the respective outputs of the two compressors will be $\frac{1}{8} \div \frac{1}{6} = \frac{3}{4}$ or 0.750. As compared with this, the ratio of the respective barometric pressures is $\frac{10}{14} = 0.714$.

Mechanically Controlled Inlet Valves for High Altitudes. It is often stated that compressors the inlet valves of which are under mechanical control are of special advantage for work at altitudes above sea-level. While there is a measure of truth in this, the possible saving is necessarily small, except at considerable elevations. The question presents itself as follows: If the valve resistance be diminished by introducing mechanical control, so that under normal conditions at sea-level the inlet air will begin to enter the cylinder a little earlier in the stroke, the volumetric capacity of the compressor is thereby increased. The loss of capacity due to resistance of the valve springs, etc., which has been assumed to be 0.75 lb. for ordinary poppet valves, is a constant, and therefore becomes proportionately of greater and greater consequence as the altitude increases, because its ratio to the diminishing atmospheric pressure goes on increasing. The percentage of saving obtained by eliminating the spring resist-

ance, though small at or near sea-level, therefore becomes a matter of importance at great elevations; and the inlet valve which presents the smallest resistance to the entrance of the air into the cylinder will be the most economical for service in high mountain regions.

Stage Compression at High Altitudes. According to the statement already made, the greater the altitude above sea-level the greater is the difference between the delivery pressure and atmospheric pressure; that is, the ratio of compression is greater. In Chap. V the effect of clearance in the air cylinder was discussed, and it is evident that the percentage loss from this cause increases with the altitude, because the piston must advance farther before the clearance air has been re-expanded to a pressure below the diminished atmospheric pressure. Even if it be questioned whether it is worth while at sea-level to adopt stage compression for the ordinary pressures used in mining and tunnelling, the case is materially altered at high altitudes. For example, if it be desired to produce a gage pressure of 75 lbs. at 5,000 ft. elevation, corresponding to an atmospheric pressure of about 12.2 lbs., 7.15 compressions are necessary. At sea-level this number of compressions would give a gage pressure of $(14.7 \times 7.15) - 14.7 = 90.4$ lbs. So far as losses due to piston clearance are concerned, therefore, it is as reasonable to employ stage-compression for 75 lbs., at 5,000 ft. elevation, as for 90 lbs. at sea-level. In a compound compressor, too, it must be remembered that there is practically but one clearance space: that in the intake cylinder. The value of the intercooler also increases with the altitude, because, in beginning compression at an initial pressure below the normal, the greater total range of pressure through which the air must be carried involves the production of more heat. This additional heat must be effectually dealt with by the cooling arrangements, if loss from this cause is to be avoided.

Considered from both the economic and thermodynamic standpoints, there can be no question as to the value of stage compression for high altitudes. There is not only a decrease in output and an increase in the cost of production of the air, due

to the added power required; but, as a result of these conditions, the compressor itself must be larger for a given output, and therefore its first cost will be greater than that of a compressor of the same capacity, working under normal atmospheric pressure. Hence, by introducing stage compression, a larger percentage of saving is possible at high altitudes than at sea-level.

CHAPTER XIV

EXPLOSIONS IN COMPRESSORS AND RECEIVERS *

EXPLOSIONS in air compressors and receivers occur with sufficient frequency to demand careful attention. Though they are unquestionably attributable to ignition of volatile constituents of the lubricating oil, the immediate causes leading to this combustion are not altogether clear. But, since explosions occur only in dry compressors, some light may be thrown upon the subject by considering the conditions affecting the use of the lubricant. In Chap. V attention was called to the fact that, if the cylinder temperature of a dry compressor rises too high, not only does proper lubrication become difficult, but the oil itself may be decomposed. Ignition unattended by actual explosion is probably frequent; the discharge pipe near the compressor sometimes becomes red-hot, and ignition has even extended into the receiver without producing a destructive explosion. The discharge-valve chests and passages, and the pipe leading from the compressor to receiver, often contain a black, sooty residue from decomposition of the lubricant. But, on passing with the compressed air into the receiver, the volatile constituents of the oil thus liberated would make a mixture of air and gas capable of producing an explosion. The extreme violence of such explosions is probably due in part to the high air pressure in the valve passages, discharge pipe, and receiver, since in high pressure air combustion is more active than in air at atmospheric pressure.

As a number of the recorded compressor explosions have occurred at collieries, the possible effects of the presence of

* In connection with the revision of this chapter I have received valuable criticisms and suggestions from my friend Mr C. M Spalding, Mechanical Engineer with the General Electric Co. This help I desire gratefully to acknowledge

coal dust in the intake air of the compressor have been considered. Such a deposit in the valve passages, together with the sooty residue from decomposition of the oil, might produce a condition favorable to an explosion. A spark caused by the friction of the compressor piston, if working dry, or, the continual passage of air at a high temperature over the carbonaceous deposit, might produce spontaneous combustion, and ignite the inflammable mixture of oil-vapor and air.* However, there are enough cases where explosions have occurred at mines and works other than collieries to prove that explosions are not necessarily dependent upon the presence of coal dust in the intake air. When the compressor is improperly situated in a room close to the boilers, some coal dust might be present in the air; but, though possibly assisting in the explosion, the quantity could hardly be large enough to produce by itself the observed results.

The primary cause of compressor explosions is undoubtedly to be found in the working conditions prevailing in the cylinder. In single-stage dry compressors very high temperatures are often reached, due to poor design of the air cylinder, or running too fast (as when the compressor is too small for its work), or attempting to produce too high a pressure. The temperature of the discharge air from a single-stage compressor is found by the formula given in Chap X,

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n}=0.29}$$

in which: T and P are the absolute initial temperature and pressure of the intake air; T′ and P′, the absolute final temperature and pressure; and n, the constant 1.41. Under normal conditions near sea-level, when the temperature of the atmosphere is 70° F., P = 14 lbs., and the gage pressure at discharge, 80 lbs., the final temperature would be:

$$T' = 70+459^{\circ}\left(\frac{80+14}{14}\right)^{0.29} = 917^{\circ} \text{ F. absolute, or } 458^{\circ} \text{ F. by}$$

the thermometer.

* T. G. Lees *Trans. Federated Inst. Mining Engineers*, Vol. XIV, p. 568.

In using this formula, the compression is supposed to be purely adiabatic, no account being taken of loss of heat by radiation or of any cooling effect from the water-jackets. Little heat can in any case be abstracted by the jackets of a single-stage compressor. Air is a poor conductor, and the volume in the cylinder is not long enough under the influence of the jackets to be much affected by them. In compressors of this type the chief office of the jackets is to keep down the temperature of the cylinder walls and prevent the lubricating oil from being carbonized. It is probable that in a single-stage dry compressor, even if well designed and in good order, the discharge temperature generally ranges from 375° to 425° F., and may go higher.

In view of these considerations the quality of the lubricating oil used in the air cylinder, and especially its flashing- and ignition-points, are matters of importance.* The flashing-point of ordinary cylinder oil may be taken as from 330°–425° F. "An average of determinations on 40 samples of heavy oils having an average flash-point of 360° F., gave an average burning-point of 398° F. High flash-test cylinder oils, from 500°–560° F., gave burning-points of 600°–630° F."† Common lubricating oils flash at about 250° F., and kerosene, sometimes carelessly used for cleaning valves, at 150° F. or below. In the case of one explosion the flash-point of the cylinder oil was found to be only 295° F.‡ It would appear, therefore, that an explosion in a compressor cylinder, directly traceable to decomposition of the lubricant, is possible under normal conditions only when inferior, light mineral oils are used.

To produce an explosion there must be a sufficient increase of temperature to cause ignition of the lubricating oil or other combustible. In endeavoring to account for abnormal compressor temperatures, different theories have been advanced.

* The flashing-point of oil is the lowest temperature at which it gives off combustible vapors in sufficient quantity to be ignited by contact with flame. The ignition-point is the temperature to which the vapors must be raised in order to continue to burn.

† Alex. M. Gow, *Engineering News*, March 2d, 1905, p. 221.

‡ John Morison, *Trans. North of England Inst. Min. Engs.*, Vol. XXXVIII, p. 6.

Some engineers have held that high cylinder temperatures may result from leakage of delivery valves, or past the piston; the argument being that the hot, high-pressure leakage air raises the initial temperature of the cylinderful of air to be compressed on the next stroke, so that the final temperature becomes abnormally high. It would appear that this reasoning does not take account of the fall in temperature due to re-expansion of the leakage and clearance air behind the piston on the intake stroke. As it may fairly be assumed that the compression cycle is approximately adiabatic, the fall in temperature of the re-expanded air (disregarding the heating effect of the hot cylinder surfaces) would nearly correspond to the original rise of temperature due to the compression of this air. This theory, therefore, does not seem tenable. Other causes may possibly exist, but we have no definite knowledge, in fact, as to what does take place in an air cylinder working hot enough to produce an explosion. In the absence of exact data, some light on the subject may be obtained from a study of the following examples:

Examples of Explosions. An explosion which took place in one of the receivers of a compressor at the Clifton Colliery, England, attracted much attention, and is so instructive that some of the details are given here.* The air from the compressor passed to a series of 3 large receivers, the first being 7 ft. diameter by 40 ft. long. While running apparently under normal conditions the safety valves of the receivers suddenly began blowing off with a deafening roar. Flames several feet high issued at great pressure from the safety valves, and sparks were blown out at the joints of the 8-in. pipe leading from the compressor to the first receiver. The air main near this receiver was nearly red-hot. That the receivers did not burst was thought to be due to the relief afforded by the 4 safety valves—2 on the first receiver and 1 on each of the others—and to the fact that the underground engines driven by compressed air continued running for some minutes after the compressor was stopped. On examining the first receiver, after it had cooled, it was found

* T. G. Lees, *Trans. Federated Inst. Mining Engineers*, Vol. XIV, pp. 555–559.

that, just below the point where the air entered from the compressor, a mass of black carbonaceous matter had been deposited, from 1½ to 2 ins. thick and 6 sq. ft. in area. On analysis this showed: volatile matter, 55.8%, fixed carbon, 37.3%, and ash, 6.9%. The material was charred and had the appearance of hard vulcanite. A thin coating was noticed on the sides of the receiver (though only near the inlet pipe) and also in the pipe itself. The other two receivers were free from deposit. A carbonaceous coating, to a thickness of ¼ in. was found on the discharge valves and passages. The cylinder and piston surfaces were not dry and, though they showed signs of excessive heat, were uninjured.

The gage pressure was usually 60 lbs., which, with adiabatic compression, corresponds theoretically to a final temperature of 405° F., the temperature of the intake air from the engine-house being 80°. The lubricating oil used was guaranteed to have a flash-point of 554°, and ignition-point of 600° F. As the cylinders were water-jacketed, the discharge air should not, in regular working, reach these temperatures; in fact, readings previously taken from a thermometer in the outlet pipe showed that it usually registered about 350° F. It is significant, however, that on a previous occasion the mercury rose above 500°, and as the thermometer tube burst, the temperature at the time of the explosion was not known. Afterward a pyrometer was fixed on the outlet pipe close to the discharge valves, and the temperature was found to range generally from 400°–420° F., varying with the speed of the engine and the air pressure produced. Even with these temperatures, high as they are, it would seem impossible that ignition of the lubricating oil used could take place. It is evident that an unusual increase of temperature in the air cylinders must be accounted for, but no satisfactory explanation of this explosion has been offered.

A violent explosion occurred in the discharge pipe of a 4-stage Laidlaw-Dunn-Gordon compressor, at a plant of the H. C. Frick Coke Co., Brownfield, Pa., which was furnishing air at 1,000 lbs. pressure for air locomotives. The compressor

was not damaged, though a large hole was blown in the pipe. It was thought that too much cylinder oil had been used, the record showing the consumption during the 5 months preceding the explosion to be 12 gals. per month. The average for the preceding year was 52.2 gals., but the reduction, great as it was, seemed to have been insufficient.*

Evidence as to another cause of trouble was obtained when a second explosion in the same compressor took place two years later. A recording thermometer, which had been installed in the discharge pipe close to the compressor, generally registered from 230°-250° F., seldom exceeding 240°. A fusible plug, designed to blow out at between 325° and 350° F., was also set in the discharge pipe near the compressor. The monthly consumption of oil was further reduced to only 3.72 gals., a solution of castile soap and water being used almost exclusively for internal lubrication, with very good results.

Previous to the second explosion, the compressor had been running normally. The day before, the maximum temperature was 240° F., the thermometer generally registering between 190° and 230°. On the day of the explosion, the temperature reached 250° between 8 and 9 A.M. By 11 A.M. it was evident that something was wrong, the temperature almost reaching 270° at 11.15. Investigation showed that the fourth-stage discharge valves were out of order, but the engineer thought that by careful running he could finish the day. He held the temperature between 250° and 265° until 2.50 P.M., when the explosion occurred; the chart of the recording thermometer then showing 270°, followed by a high peak in the curve. Coincident with the explosion, the fusible plug melted and blew out, releasing the pressure and checking the temperature at 620°. The compressor, which was uninjured, was stopped, and a new plug put in, taking about 15 mins., during which time the temperature dropped to 245°. On starting again (in doing which the engineer assumed an unnecessary risk) the temperature rose to 270°, before the compressor was shut down at 4.10 P.M.

* It is probable that gummed oil and carbonaceous deposit had accumulated liberally wherever it could lodge in the interior of the compressor.—R. P.

New valves and seats were put in, and on starting again 2 days later the temperature ranged from 220°–240°.*

This explosion was ascribed by the management to "churning" of the air, due to leaky discharge valves, which allowed the high-pressure air to re-enter the fourth-stage cylinder. It is possible, however, that the valves of the cylinder were not working at all when the explosion took place, in which case the compressor became temporarily a 3-stage machine. If this be true, in compressing to 1,000 lbs. in three cylinders, the compressor was working under conditions for which it was not designed, and for which the cooling arrangements of the three remaining cylinders were inadequate. This explanation does not appear to be unreasonable.

During the construction of the New York Aqueduct a fire occurred in a compressor receiver at one of the shafts. The air pressure was 80–90 lbs., and the receiver, set outside of the engine-house, was exposed to the hot sun. Part of the discharge pipe leading to the receiver became red-hot. On stopping the compressor and cooling down the receiver, the entire inner surface of the latter was found to be coated with carbonaceous matter at least ⅛ in. thick. Further investigation brought out the fact that the poppet discharge valves had sometimes occasioned trouble by sticking, and the engineer had been in the habit of using a squirt-can of kerosene to cut the gummy material clogging them. As the kerosene had a low flash-point, it was quickly vaporized, and when the cylinder temperature reached a sufficiently high point the explosion took place.

In a case at Butte, Mont., two duplex compressors, with air cylinders respectively of 32¼ by 60 ins. and 24¼ by 48 ins., and running at 50 revs. per min., were forcing air at 80 lbs. pressure through a single 8-in. pipe. As somewhat over 1,200 cu.ft. of compressed air per min. were being produced, the velocity of flow would be nearly 3,500 ft. per min., or 58 ft. per sec. It had been noticed several times that a portion of the discharge pipe close to the compressor became red-hot. In

* Abstracted from a paper in *Mines and Minerals*, Vol. XXXII, p. 651, by William L. Affelder, Gen. Mgr. Bulger Block Coal Co., Bulger, Pa.

the pipe between the compressors and receivers were several sharp bends, which increased the friction due to the rapid flow of the air. The receivers were always extremely hot. On one occasion the shaft timbering, 40 or 50 ft. below the shaft mouth, took fire from the hot air pipe.

Although the observed results of this explosion were localized in the discharge pipe, it is probable that oil was first vaporized either in the cylinder or when the compressed air was passing through the delivery valves; that a portion of it became hot enough to ignite, and in turn ignited an accumulation of oil vapor in the discharge pipe to the receiver.

It seems necessary to hold that the primary cause of explosion is to be looked for in the cylinder, not in the discharge pipe or receiver. That is, it is reasonable to assume that the conditions leading to explosion are initiated at the point of maximum pressure (and therefore of maximum temperature), which is towards the end of the stroke and while the air is passing through the discharge valves. If this temperature is high enough, vaporization of some of the lubricating oil will occur, followed by ignition, which might extend into the mixture of air and oil vapor in the discharge pipe.

Foul or poisonous gases may result from ignition of the lubricant in compressors or receivers, not necessarily followed by actual explosion. In an article in the *Trans. Amer. Inst. Min. Engs.*, Vol. XXXIV, p. 158, an instance is noted of combustion in an air pipe and receiver. The compressed air was being used in an imperfectly ventilated upraise in a mine, 1,200 ft. from the compressor, and two men lost their lives, while four others barely escaped asphyxiation.

Other more or less similar cases are familiar to most miners, where foul air from the exhaust of machine drills has been observed; sometimes merely disagreeable, though often actively deleterious. The use of poor cylinder oil is probably responsible for this, as its lighter constituents may begin to volatilize and burn at a normal working temperature. Even if not actually fried on the hot metal surfaces, a low-grade oil will undergo a slow combustion or oxidation, which may produce enough

carbon monoxide to raise materially the percentage of that poisonous gas in the confined atmosphere of working places of mines.

Mode of Using Lubricant for Air Cylinders of compressors. Sight-feed lubricators, as commonly employed for steam cylinders, are best. On the Clifton Colliery compressor, mentioned above, ordinary oil-cups were used, holding about ⅛ pint; they were filled 4 times per day of 10 hours. With these oil-cups, if improperly adjusted, it would be possible for all the oil to be sucked into the cylinder within a few strokes after being filled. Such a result might be inferred, indeed, in this case, because of the large quantity of carbonaceous matter—oil, coal dust, etc.—found in and around the discharge valves and in the receiver. The oil feed should be carefully regulated, and a smaller quantity used in an air cylinder than a steam cylinder of the same size—say, one-third as much. Excess of oil increases the tendency to gum the valves. For stage compressors of ordinary size, 1 drop of good cylinder oil every 4–5 minutes is sufficient.

The periodical use of soap and water (soap-suds) is recommended for any compressor that cannot be shut down at short intervals for overhauling. It is fed into the air cylinder through an oil-cup, say during one day per week. Or it may be forced in by an oil-pump, with which the compressor should be provided. Soap and water is a poor lubricant, and must be used more freely than oil, but it is effectual in cleansing the cylinder, valves, and ports from carbonaceous or gummy matter. If the compressor is to be stopped, as at the end of a shift, the feeding of soap and water should be discontinued some time before shutting down, and the oil-feed resumed, to avoid formation of rust. Every compressor should be overhauled from time to time, and thoroughly cleaned in all parts, especially around the valves and passages, capable of furnishing a lodgment for oil or partly oxidized carbonaceous material.

Precautions for Preventing Explosions: (1) Always inclose the inlet valves in a cold-air box, connecting with the outside air, to avoid taking air from the hot engine-room. This conduces to economy in working, and by keeping down the final

temperature tends to prevent decomposition of the oil. (2) The largest possible area of cylinder surface should be water-jacketed, including the cylinder heads. A liberal supply of the coldest water obtainable should be used for the jackets. The advantages in this respect of employing stage compression, with large inter- and aftercoolers, are undoubted. (3) Use only the best cylinder oil, with high flash- and ignition-points and in as small quantity as is consistent with proper lubrication. (4) Keep the valves clean. In the design of the compressor there should be no recesses or pockets, around the valves or passages, where oil could accumulate. (5) Never introduce kerosene into the cylinder for cleaning the valves while the compressor is running. (6) Arrange the air intake so that coal dust will not be drawn into the cylinder with the inlet air. (7) Place a thermometer in the discharge pipe, close to the cylinder, so that the engineer can watch the temperature, and stop or slow down the compressor if the temperature of the discharge air rises too high. A continuously-recording thermometer is to be recommended.

Conclusions. Though compressor explosions are not uncommon, it is undoubtedly true that, while mixtures in certain proportions of air and oil vapor are explosive, oil is often burnt in the cylinders without causing a destructive explosion. This is proved by the frequent presence in air cylinders of sooty, carbonaceous deposits. The theories aiming to account for the observed phenomena by placing the responsibility entirely on leakage of discharge valves, or on "churning" of the air back and forth in the cylinder, due to sticking of the valves, are not conclusive nor satisfactory for the reasons stated on p. 201 regarding the adiabatic compression cycle. The whole subject is at present obscure.

Before attempting to formulate conclusions, it would be desirable to secure more data. Definite knowledge of the conditions leading to explosions can be obtained by making laboratory investigations under controlled conditions, and applied to the circumstances which have been assumed to cause explosions. In this way, it could be determined to what extent, if any, the cylinder temperature is raised by leaky discharge

valves, or by "churning" of the air. The effect of small particles of relatively non-conducting material, as carbon (coal dust) or lint, drawn into the cylinder with the intake air, could also be investigated. Such carbonaceous points might become incandescent, due to heating from small jets of flame from burning oil, and remain so long enough to ignite larger volumes of mixed air and oil vapor. It may be suggested that research leading to fuller knowledge of this subject might be carried out in the mechanical engineering laboratories of a university, or at the works of some compressor builder.

CHAPTER XV

AIR COMPRESSION BY THE DIRECT ACTION OF FALLING WATER

Principle. When air in small bubbles is intimately mixed with water, the water breaks into foam, through which the bubbles tend to rise and escape. But if the mixed air and water be drawn downward by a strong falling current, as in a vertical pipe, the air is compressed. Then if, after reaching the depth and head of water-column necessary to produce the compression desired, the direction of flow be changed to the horizontal and the velocity diminished, the bubbles of compressed air will be liberated and may be collected in a suitable chamber. The air pressure in this chamber corresponds to the effective head of water, that is, its depth below the level at the outflow or tail-race. Thus, in this method of compression, no piston, valves, nor other moving parts, are used.

As the bubbles are minute and thoroughly disseminated through the water during its descent, the total cooling surface is very large and isothermal compression results. The moisture-carrying capacity of the air is therefore smaller than if it were compressed adiabatically. During compression the percentage of moisture in each globule of air increases until the point of saturation is reached; on further compression, the excess moisture is deposited, so that when re-expanded the air is relatively dry. This method was first tested on a working scale about 1878, by J. P. Frizell, of Boston, Mass.*

Magog Plant. In 1896, the Taylor Hydraulic Air Compressing Co., of Montreal, erected a plant for the Dominion

* *Proceedings*, Institution of Civil Engineers, London, Vol. LXIII, p. 347.

Cotton Mills, Magog, Province of Quebec.* In a 128-ft. shaft (Fig. 114) was erected a vertical compressing pipe *a*, 3 ft. 8½ in. diameter, the lower part increasing to 4 ft. 8 in., and made of $\frac{5}{16}$-in. steel-plate. This pipe passes through the bottom of a receiving chamber *b*, 12 ft. diameter by 12 ft. high, to which water is conducted from a head-race. Water flows into and fills the pipe, which extends nearly to the bottom of the shaft. Through a series of small feed pipes, air is drawn with the water into the top of the main pipe and is compressed while being carried down the shaft. The compressed air collects in a chamber *c*, while the water is returned to a tailrace near the top. The difference of level between intake and tailrace is about 22 ft., which produces the requisite speed of flow. Into the top of the vertical pipe *a* is inserted a telescoping section of pipe *d* (Fig. 115), carrying a bell-mouth *e* and headpiece *f*, terminating below in an inverted conoid *g*. Between *e* and *g* is an annular opening, through which the water enters the compressing pipe. The headpiece carries thirty 2-in. pipes *h*,*h*, 4 ft. long, open at the top and closed at the bottom Into each of these pipes are screwed 32 short horizontal $\frac{3}{8}$-in. pipes *i*,*i*, all directed into the annular opening between *e* and *g*. As the entering water passes among the small pipes air is entrained, carried down the main pipe in the form of bubbles, and is thus compressed.

Near the bottom of the shaft the compressing pipe enters the "separating" chamber *c*, 17 ft. diameter and 12 ft. high, open below and supported upon legs which raise it 16 ins. above the shaft bottom. Within this chamber is a conoidal "disperser" *j*, 12 ft. diameter. Below is an apron *l*, 5 ft. wide. When the water, charged with air bubbles, reaches the disperser it is first directed outwards, then deflected by the apron toward the center, and finally escapes through the open bottom of the separating tank into the return column. During this process of travel the compressed air separates from the water, most

* The following description is based on an article in the *Canadian Engineer*, March, 1897, and information furnished to the author by the builders. See also *Eng. and Mining Jour.*, Dec. 26th, 1896, p. 606, and *Railway and Engineering Review*, Sept. 17th, 1898, p. 513.

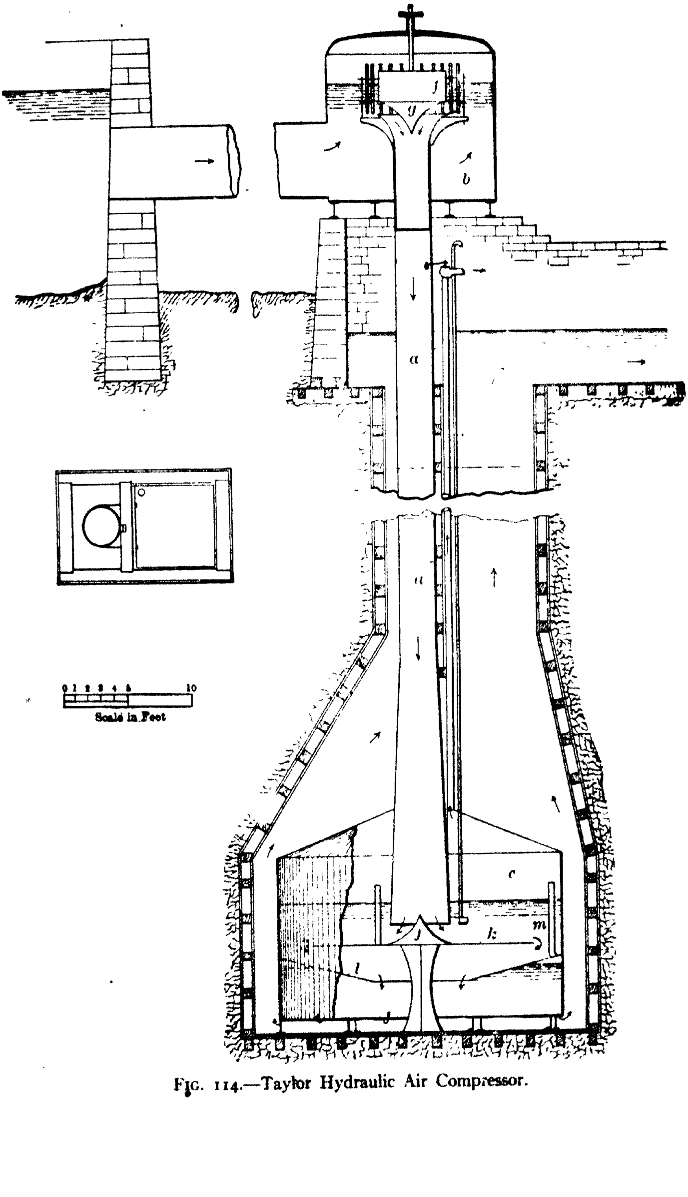


FIG. 114.—Taylor Hydraulic Air Compressor.

of it collecting in the upper part of chamber *c*. Part of the air is not liberated at once, but collects in the annular space under the apron, and joins the main body of air through the pipe *m*. The pressure in the air chamber is due to the height of the re-

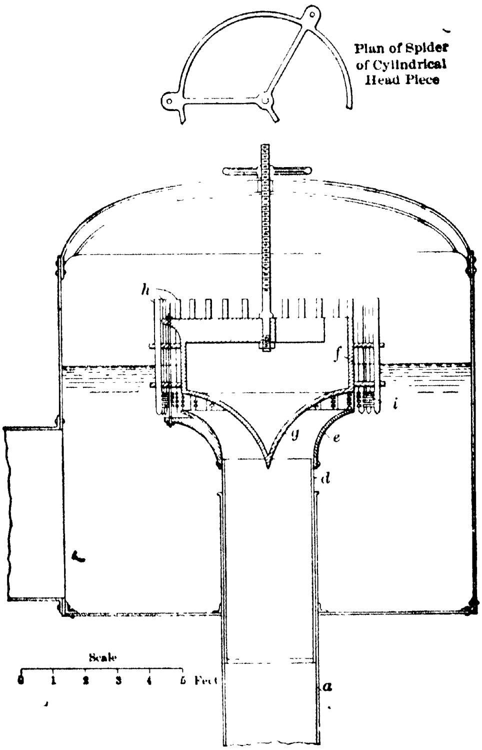


FIG. 115.

turn water column in the shaft. The air is drawn off through the air main, alongside of the water column *a*. As the air bubbles are surrounded by cold water, perfect isothermal compression is attained, with its corresponding advantages in minimizing the amount of moisture carried off in the air.

TABLE XIV.—TESTS ON THE MAGOG PLANT.*

No. of Test.	Water Discharged, Cu.ft per Min	Available Head, Ft.	Available Horse-Power.	Air Delivered, Cu.ft per Min. at Atmos. Pressure.	Air Pressure, Lbs. per Sq. in.	Actual Horse-power of Compressor.	Efficiency, Per Cent.
1	6122	21.4	247.7	1377	52	132.5	53.5
2	5504	21.9	228.0	1363	52	131.0	57.5
3	4005	22.3	168.9	1095	52	107.3	62.4
4	7662	21.1	305.9	1616	52	155.4	50.8
5	6312	21.7	260.0	1506	52	144.8	55.7
6	7494	21.2	290.8	1560	52	150.2	50.1

Temperatures during tests external air, 75-80°; water, 75.2-80°; compressed air, 75.2-80°.

The parts were incorrectly proportioned in this first installation, and the efficiency could be increased by using a larger air chamber, to prevent air from going to waste.

The theory of this mode of compression is as follows: The combined specific gravity of the mixture of air and water in the compressing pipe is less than that of the water in the return column. Therefore, the head required to overcome friction and to produce flow must be greater than if the apparatus were merely an inverted siphon, and as the difference in weight increases with depth (and air pressure produced) the motive head, or difference in level between the surfaces of water at inlet and in tailrace, must be correspondingly increased.

Kootenay Plant. In 1898–1900 another plant was built for the Kootenay Air Supply Co., Ainsworth, B. C. The topographical conditions are such that a high head is obtained without sinking a deep shaft. From a dam the water is carried in a wooden-stave pipe, 5 ft. diameter and 1,354 ft. long, over a short trestle, built against the side of a gorge, to the receiving tank. The latter, 17 ft. diameter by 20 ft. high, is placed on a wooden tower, 110 ft. high (Fig. 116). From the tank the pressure pipe, 33 ins. diameter, descends to the ground level and then down a shaft 105 ft. deep.† After compressing the air

* Tests made by Prof C. H. McLeod, of McGill University, August, 1896. Published in *Eng. and Min. Journal*, December 26, 1896, p. 606.

† *Canadian Electrical News*, September, 1898, p. 176.

the water returns up the shaft to the tailrace at the creek level. Fig. 117 shows the details of the receiving chamber at the bottom of the shaft.

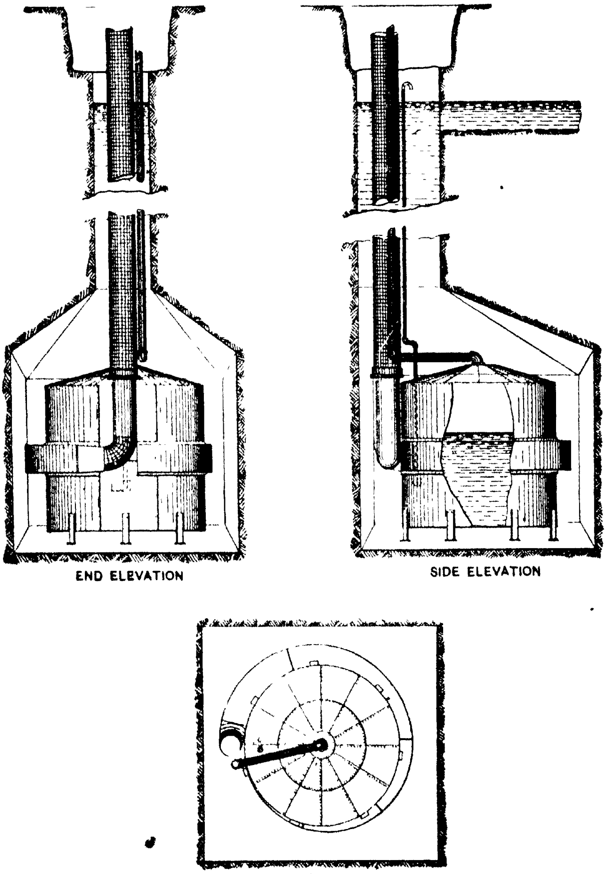


FIG. 117.—Hydraulic Air-Compressor at Kootenay.

The effective compressing head is 107 ft., the total height of the intake pipe being over 200 ft. This produces a high velocity of flow and large delivery of compressed air. The compressed air main, 9 ins. diameter, is 2 miles long, carrying from 4,200 to 4,600 cu.ft. of free air per min. Branch service pipes convey the air to neighboring mines, where it is used for rock-drills and other machinery. On the basis of 600 H.P., represented by the volume and pressure of the air, the cost of the entire plant, including pipe lines, was about $100 per horse-power.

Victoria Plant was completed in 1906 at the Victoria Copper Mine, Rockland, Ontonagon Co., Mich. The local conditions led to a novel mode of installation. The water is conducted from a dam on the Ontonagon River through a 4,700-ft. canal, furnishing a head at the terminal forebay of 72 ft. above the river-level. Three independent units are built side by side at 19-ft. centers in a vertical shaft 340 ft. deep. In the original design, the subdivision of the air, as admitted at the intake head (Fig. 118) was carried farther than in either of the plants described above, by inserting 1,800 $\frac{3}{8}$-in. horizontal feed pipes, in the series of larger vertical pipes encircling the inverted cone.

After the plant was put in operation, serious trouble was experienced by the freezing up of the small pipes of the intake heads, due to the severe winter climate of the region. This led to the removal of the heads, the water being allowed simply to flow into the top of the compression pipes. The breaking up and agitation of the mass of water, in changing its direction of flow from the forebay into the compressing pipes, entrained the air quite efficiently, and it is stated that the capacity of the plant, in cubic feet of free air compressed per minute, is practically the same as when the intake heads were in use.

The compressing pipes are 5 ft. diameter, lined with concrete, and separating cones and dispersers, also of iron and concrete, are built in a chamber at the bottom. In this chamber, 281 ft. long and 18 ft. by 21 ft. average cross-section, the compressed air is trapped and thence drawn off through a 24-in. main. The compressing water, flowing down the intake pipes,

stands normally at a level about 14½ ft. below the roof of the chamber, thus leaving an air capacity of about 80,000 cu.ft. Connected with the end of the air chamber is an inclined shaft, 270 ft. in vertical depth, through which the water returns to the surface. The tailrace from this shaft is 72 ft. below the level of the intake, this height measuring the motive head producing the flow of water. Thus the air in the underground chamber is under a pressure due to 270 ft. head of water, or 118 lbs. sq. in.

For regulating the operation of the original plant a pipe passed from the air chamber up the compressing shaft to the surface, whence branches were led to the intake heads. The compressed air conveyed in this regulating pipe operated a device connected with each intake head, whereby the latter was automatically raised above the water-level in the receiving tanks whenever the air pressure exceeded the normal, thus stopping the flow of air through the feed pipes. A 12-in. blow-off pipe passes from the water-level in the air chamber to the mouth of the inclined shaft carrying the return water column. If air to the full compressor capacity is drawn off, the water-level in the air chamber rises as the air pressure falls, thus sealing the lower end of the blow-off pipe; then, when the consumption of air decreases, the pressure in the chamber rises, depressing the water-level until the blow-off orifice is uncovered, when more air is blown off. Thus the working pressure is maintained within quite narrow limits. The great size of the air chamber—corresponding to the receiver of an ordinary air-compressor—gives a large storage capacity.

When all 3 compressing units are in operation, with a total capacity of from 34,000 to 36,000 cu.ft. of free air per min., about 70,000 cu.ft. of air per min. may be drawn off for a period of 18 minutes, without causing a drop in pressure of more than 5 lbs. For each unit, the output ranges from 9,000 to 12,000 cu.ft. per min., and the volume of water used from 12,700 to 14,800 cu.ft. Tests made on a single intake head in May, 1906, by Prof. F. W. Sperr, gave the following results:*

* For further details see article by D. E. Woodbridge, *Eng. & Min. Jour.*, Jan. 19, 1907, p. 125. Also, A. H. Rose, *Mines & Min.*, Mch., 1907, p. 346.

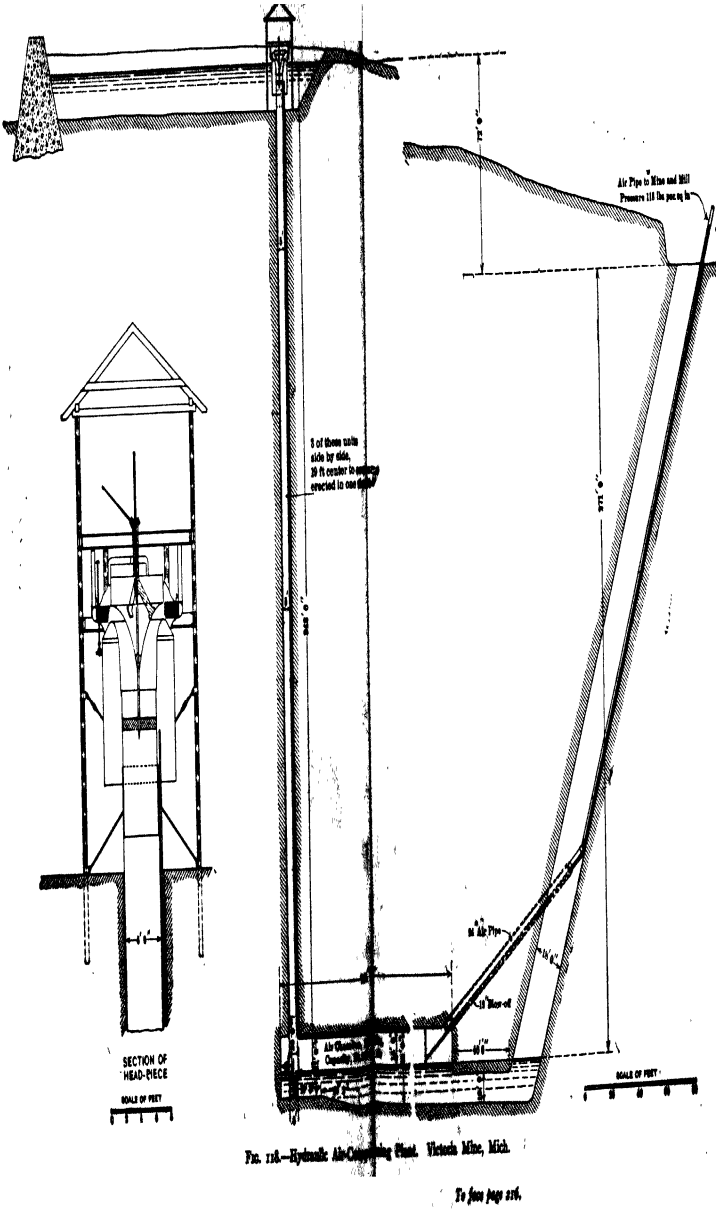


FIG. 118.—Hydraulic Air-Compressing Plant. Victoria Mine, Mich.

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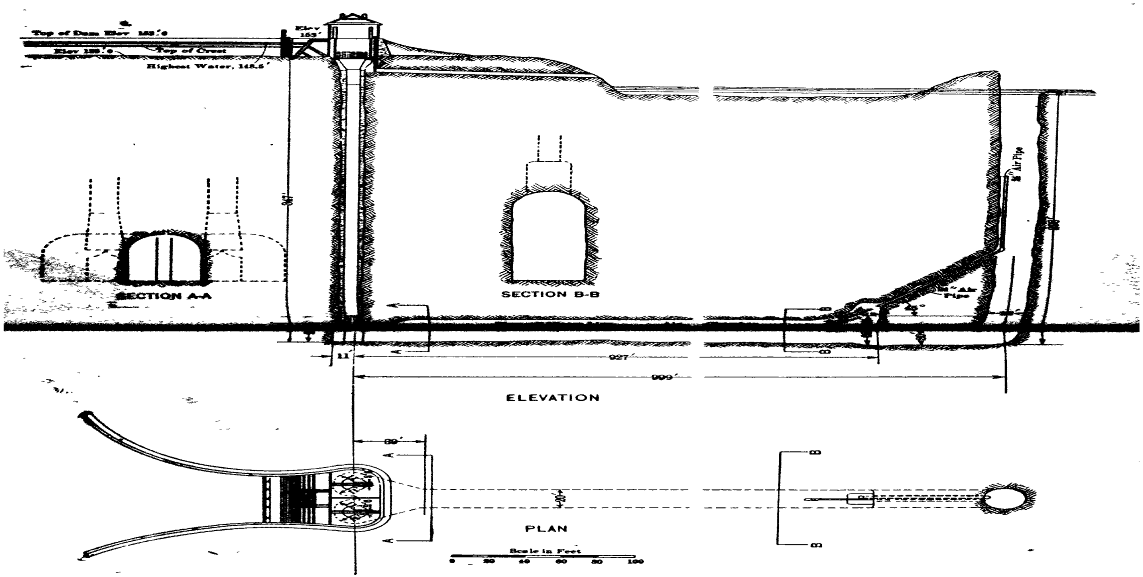

FIG. 119.—Cobalt Power Co's Plant, Plan and Elevation.

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TABLE XV.—AIR MEASUREMENTS

Sq ft	Velocity, Ft. per Sec.	Cu ft. per Min.	Absolute Pressures. Free Air, Lbs.	Absolute Pressures. Compressed Air, Lbs.	Horse-power.
4	44.09	10,580	14	128	1,430
4	49.74	11,930	14	128	1,623
4	38.50	9,238	14	128	1,248

WATER MEASUREMENTS

Flume Area	Velocity, Ft per Sec.	Cu ft per Min	Head, Ft	Horse-power	Efficiency, Per Cent.
71.75	3.033	13,057	70.5	1,741	82.17
67.03	3.684	14,870	70.0	1,961	82.27
72.16	2.936	12,710	70.6	1,700	73.50

The air is used at the Victoria Mine for general power purposes at the mine and mill, including a 500-H.P. hoisting-engine, and 7 pumps. The cost per horse-power is about $2.25 per year, including all operating expenses. Over 4,000 H.P. can be developed by the 3 compressing units.

Cobalt Power Co.'s Plant. Following a series of efficiency tests made in 1909 on a large number of steam compressors at the silver mines in the Cobalt, Ont., district (see Chap. X, p. 152–168), a Taylor compressor was built at Ragged Chutes, 9 miles from Cobalt, on the Montreal River. In a distance of 1,000 ft. there is a drop of 54 ft. From the forebay the water flows into two 16-ft. heads (Fig. 119), in each of which sixteen 14-in. vertical intake pipes are set in a horizontal disk. Below the disk the heads taper to 8 ft. $4\frac{3}{4}$ in., below which point they extend 15 ft., telescoping into the tops of $8\frac{1}{2}$-ft. diameter concreted compression shafts, 330 ft. deep. To regulate the inflow, and adjust the position of the heads to the forebay water level, the heads are suspended from 2 vertical hydraulic cylinders. The heads, with their large diameter intake pipes, are designed to prevent freezing in the severe winter climate of the region (see above, under Victoria Plant).

The water with the entrained air flows through the heads at a velocity of 15–19 ft. per second. Due to the compression of the air in the shaft, this velocity gradually diminishes, with a further reduction in the lower 40 ft. of shaft, which is flared to 12½ ft. diam. At the bottom of each shaft is a steel-plate capped concrete diverting cone (see also Fig. 118 and accompanying description).

The shafts terminate in a tunnel (air chamber), 26 ft. high, 20 ft. wide and 1,000 ft. long, in which the air is completely liberated, and which serves as a receiver. So great a length was not required for these purposes, but was adopted to utilize the total head (54 ft.) of the steam. From the tunnel, air is drawn off through a 24-in. pipe passing in an inclined riser to the vertical tail shaft. The 12-in. blowoff pipe acts in case the air pressure in the tunnel should force the water level below the roof of the outlet to the tail shaft, and so cause fluctuations of pressure. The air pressure produced is that due to the net head of water in the tail shaft; in this case, 276 ft., corresponding to 120 lbs. per sq. in.

This plant compresses 40,000 cu.ft. free air per min., corresponding to about 5,500 H.P. The air is carried through 9 miles of 20-in. pipe to Cobalt. From there branch pipes connect with the different mines, the total piping (20, 12, 6 and 3-in.) being about 21 miles. Total cost, excluding piping, about $1,000,000, or $185 per H.P. The air is sold by the company at 25 cents per 1,000 cu.ft. at 100 lbs. pressure.*

Other Plants. In the State of Washington there is a 200-H.P. plant. Head of water, 45 ft., height of compressing pipe, 260 ft., diameter, 3 ft.; volume of water, 53 cu. ft. per second; air pressure, 85 lbs.†

A small plant at Peterborough, Ont., has an 18-in. compressing pipe, in a 42-in. shaft. Depth of separating chamber below discharge level, 64 ft., air pressure, 25 lbs.

Near Norwich, Conn., on the Shetucket River, there is a large plant for general power purposes.‡

* C. H. Taylor, *Mines & Min.*, Apl., 1910, p. 532.

† *Eng. & Min. Jour.*, Apl. 27, 1901.

‡ *Compressed Air Magazine*, Apl., 1906, p. 3,980.

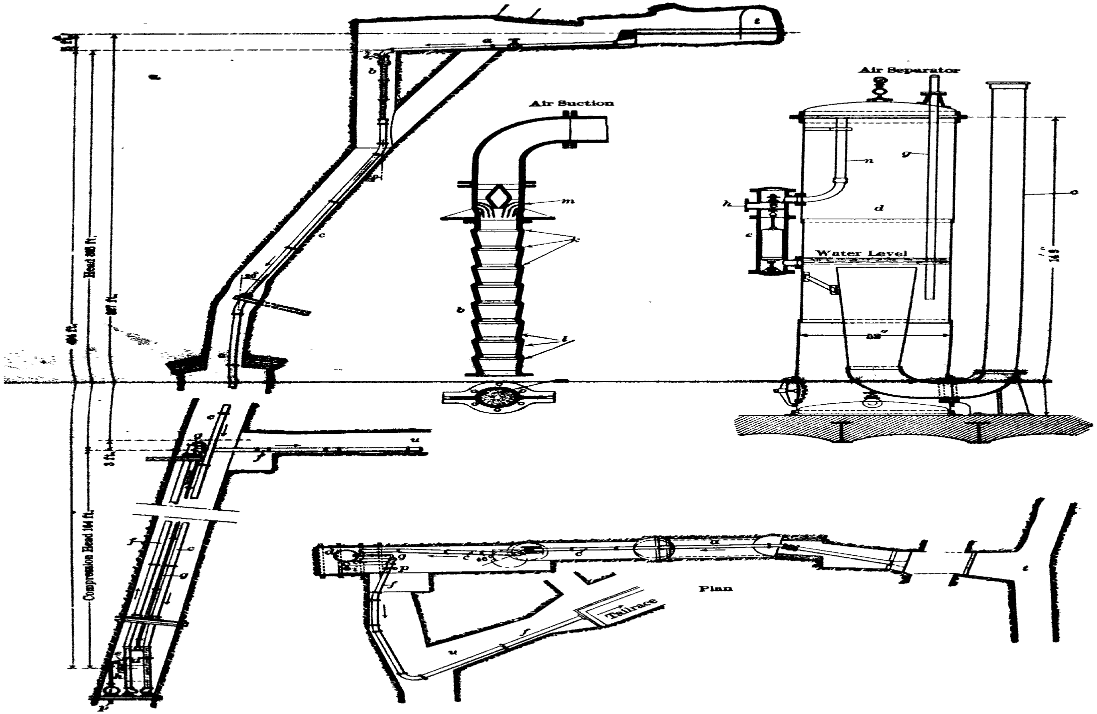

FIG. 120.—Underground Hydraulic Air Compressor, at Clausthal, Germany.

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In 1907–8 a hydraulic air compressor was installed at a silver mine at Clausthal, Germany. Fig. 120 shows the general design, with details of the intake head and compressing chamber. A flow of water in the tunnel *t* is led through an $8\frac{1}{2}$-in. cast-iron pipe *a*, to the air intake *b*, which consists of a number of flaring rings *l*, in the upper rim of each of which is a series of small holes *k*, for admitting the air. Additional inlet area is provided at the top of the intake by a nest of small curved pipes *m*. The mixed air and water pass into the $8\frac{1}{2}$-in. compression pipe *c*, 492 ft. long, laid in an inclined shaft, and discharging into the compressing chamber *d*. This chamber is 52 in. by 14 ft. 9 in. high. From a point near its top the compressed air passes through pipe *n* to the automatic check-valve *e*, and thence, by pipe *h*, to the receiver *i*. The water leaves the compressing chamber by the $8\frac{1}{2}$-in. pipe *f*, which discharges at a point 164 ft. above, into a tailrace occupying the mine level *u*. An equalizing discharge pipe *g*, from the compressing chamber, is led up the shaft, parallel to *f*, entering the latter at the level of the tailrace. Total cost of the plant is stated to be $3,750.*

The average flow of water is 792 gals. per min.; which, falling through a vertical height of 325 ft., produces theoretically 66.3 H.P. A flow of 845 gals. per min. gave 353 cu.ft. of air, at 71.2 lbs. gage. To compress 1 cu.ft. of air adiabatically to this pressure requires 0.147 H.P. and to compress 353 cu.ft., about 51.9 H.P. Since 70.5 theoretical H.P. are produced by the flow of 845 gals. per min., the efficiency is $\frac{51.9}{70.5} = 73.6\%$.

Modifications of the hydraulic compressor have been proposed: the McFarlane, described in *Eng. and Min. Jour.*, Oct. 10, 1908, and the Blakney, *Eng. & Min. Jour.*, Apl. 24, 1909. (For a general description of hydraulic compressors, see *Western Eng'g*, March, 1917.)

The first cost of hydraulic air compressors is not excessive, while the maintenance and running expenses are very low,

* Abstracted from a description by P. Bernstein, in *Glückauf*, March 14, 1908. Translation by E. K. Judd in *Eng. & Min Jour.*, August 1, 1908, p. 228 See also *Zeitschr. ver. Deutscher Ing.*, Nov. 5, 1910.

compared with those of ordinary compressors. No skilled attendance is required, and depreciation is nominal in substantially erected plants. By comparing the figures given in Tables XIV and XV, it will be seen that Victoria Plant gave a marked increase in efficiency, due to the greater motive head, and a more complete separation of the air from the water in the receiving chambers.

It has been suggested that it might be feasible to employ the system in connection with an ordinary compressor plant. That is, to produce a low air pressure by the water plant, and then to admit this air to the compressor cylinder where it would be brought to the required higher tension. In effect, this would be stage compression, in which the air would be cooled to normal temperature before entering the high-pressure cylinder.

This system of air compression is generally unsuitable for small plants, as the first cost is large.

Part Second

TRANSMISSION AND USE OF COMPRESSED AIR

CHAPTER XVI

CONVEYANCE OF COMPRESSED AIR IN PIPES

THE diameter of the pipe is of vital importance, and when proportioned properly to the volume of air, and to the distance, the transmission losses are very small compared with the other losses incident upon air compression. Transmission losses appear in two ways: as loss of power, and as loss of pressure or head.

Loss of Power. The large loss of power due to the heating of the air during compression and its subsequent cooling, has already been considered. This cooling takes place so quickly in the receiver and piping that the resulting loss is not properly chargeable to transmission. The air assumes the temperature of the surrounding atmosphere in the first few hundred feet, so that when conveyed to long distances the calculation for transmission loss may be made without regard to the effect of temperature upon the volume of the air. The power in the compressed air is due not only to its pressure, but also to its volume, in terms of cubic feet of free air. While the pressure is reduced by frictional loss in transmission, this reduction is accompanied by a proportionate increase in volume, and a certain compensation is produced. Although the pressure of the air at the motor is diminished, there is no loss in the final volume of free air. As shown below, the loss of pressure due to the conveyance

of air in pipes is small, but the actual loss of power is still smaller. The pipe itself acts in a measure like a receiver—as a reservoir of power. Much of the transmission power loss experienced in practice is due to leakage from joints and flaws in the pipe.

Loss of Pressure for short distances takes place according to the laws governing the flow of all fluids, varying directly as the length of pipe, directly as the square of the velocity, and inversely as the pipe diameter. For long distances the application of these laws becomes somewhat complex. In addition to the factors just given, it is necessary to take into account the volume and pressure of the air, and the initial and final pressures at the ends of the pipe line. In general, for a given diameter of pipe, when the volume of air discharged and its initial pressure remain constant, the loss of pressure is proportionate to the length of the pipe.

But in actual service the initial pressure and volume of discharge do not remain constant, and, in the passage of the air through the pipe, other modifying factors must be taken into account. In flowing through a long line of piping the pressure is gradually reduced by friction, while the volume is correspondingly increased. Therefore, to maintain in the pipe the flow of a given quantity of air, the volume of which is constantly increasing, the velocity also must increase, and this requires an increase of head or pressure.

The formulas commonly used assume that the loss of head is proportional to the length of pipe, so that, if a certain head be required to maintain the flow of a given quantity of air in a pipe 1,000 feet long, twice this head would suffice for a pipe 2,000 feet long. But, when the air has passed through the first thousand feet of pipe its motive head has been lost; and as the volume has thereby increased, a greater head will be necessary to maintain the flow in the second thousand feet. The ordinary formulas do not take into account the increase of volume due to this loss of head. To transmit a given volume of air at a uniform velocity and loss of pressure would require a pipe of gradually increasing area. This of course is impracticable, and if the discharge is to be kept constant in pipe of uni-

form section, both volume and velocity must increase as the pressure is reduced by friction. The loss of head in properly proportioned pipes is so small, however, that in practice the increase in volume is usually neglected.

The discharge capacity of piping is not proportional to the cross-sectional area alone. Although the periphery is directly proportional to the diameter, the interior surface resistance is greater in a small than in a large pipe, because the ratio of perimeter to area is greater. To pass a given volume of air a 1-in. pipe of given length requires over 3 times as much head as a 2-in. pipe of the same length. The character of the pipe also, and the condition of its inner surface, have much to do with the frictional resistance. The irregularities incident upon coupling together the lengths of pipe also increase friction. As the influences by which the values of some of these factors may be modified are not fully understood, the results obtained from formulas are only approximately correct.

D'Arcy's Formula, as adapted to compressed-air transmission, is:

$$D = c\sqrt{\frac{d^5(p_1 - p_2)}{w_1 l}}, \text{ or } D = \frac{c\sqrt{d^5}}{\sqrt{l}} \times \sqrt{\frac{p_1 - p_2}{w_1}}$$

in which:

D = the volume of compressed air, cubic feet per minute, discharged at the final pressure; c = a coefficient varying with the pipe diameter, as determined by experiment; d = nominal diameter of pipe, inches;* l = length of pipe, ft.; p_1 and p_2 = initial and final gage presssures, lbs. per sq. in.; w_1 = density of the air, or its weight in lbs. per cu.ft., at pressure p_1.

The formula in its factored form (see above) is convenient for use. Table XVI gives the values of c, d^5, and $c\sqrt{d^5}$.

Table XVII gives the value of w_1 for initial gage pressures up to 100 lbs. per sq. in., Table XVIII the values of $\sqrt{\frac{p_1 - p_2}{w_1}}$ for terminal pressures of 20–100 lbs., and pressure losses of 1–10

* The actual diameters of wrought-iron pipe are not the same for all sizes as the nominal diameters. This difference is small except in the 1¼-in. and 1½-in. sizes, the actual diameters of which are 1.38 ins. and 1.61 ins. respectively.

lbs. Intermediate values are obtained by interpolation. No allowance is made for pipe leakage, nor for incidental friction due to bends in the pipe (Table XXII).

TABLE XVI

Diameter of Pipe, Inches	Values of c	Fifth Powers of d	Values of $c\sqrt{d^5}$
1	45.3	1	45.3
2	52.6	32	297
3	56.5	243	876
4	58.0	1,024	1,856
5	59.0	3,125	3,298
6	59.8	7,776	5,273
7	60.3	16,807	7,817
8	60.7	32,768	10,988
9	61.0	59,049	14,812
10	61.2	100,000	19,480
11	61.4	161,051	24,800
12	61.6	248,832	30,926

* TABLE XVII

Gage Pressure, Lbs.	w_1	$\sqrt{w_1}$	Gage Pressure, Lbs	w_1	$\sqrt{w_1}$
0	0.0761	0.276	55	0.3607	0.600
5	0.1020	0.319	60	0.3866	0.622
10	0.1278	0.358	65	0.4125	0.642
15	0.1537	0.392	70	0.4383	0.662
20	0.1796	0.424	75	0.4642	0.681
25	0.2055	0.453	80	0.4901	0.700
30	0.2313	0.481	85	0.5160	0.718
35	0.2572	0.507	90	0.5418	0.736
40	0.2831	0.532	95	0.5677	0.753
45	0.3090	0.556	100	0.5936	0.770
50	0.3348	0.578			

Example: Given a 5-in. pipe, 2,500 ft. long; how many cu.ft. of air per min. at 70 lbs. initial pressure can be transmitted, with a loss of pressure of 3 lbs.?

* Reproduced by permission from *Compressed Air*, Feb., 1898, pp. 374-376.

From Table XVI, $c\sqrt{d^5}=3{,}298$; from Table XVIII, $\sqrt{\frac{p_1-p_2}{w_1}}$ $=2.615$ and $\sqrt{l}=50$. Substituting in the formula:

$$D=\frac{3{,}298}{50}\times 2.615=172.5 \text{ cu.ft. compressed air per min.}$$

Volumes of compressed air may be converted into free air by multiplying by the absolute pressure in atmospheres (1 atmos. = 14.7 lbs.). Thus, 100 cu.ft. of air at 80 lbs. gage, or 94.7 absolute pressure, correspond to 644 cu.ft. of free air, at sea-level. Table XIII gives the air pressures in lbs. per sq.in. for altitudes to 15,000 ft., with the corresponding barometric readings.

Graphic Solution of D'Arcy's Formula (C. W. Crispell, *Trans.* Am. Inst. Min. Engs., Vol. LVIII, p. 97.) The following problems may arise (see Fig. 121):

1. To find the diameter of pipe; given the volume of compressed air, length of pipe, initial pressure and maximum drop in pressure.

With a straight-edge, join the scales marked length of pipe and cu.ft. of compressed air, and note the intersection on axis *A*. Join the initial pressure with the drop in pressure, and note intersection on axis *B*. A line joining these two points of intersection will cut scale No. 3 at the required pipe diameter.

2. To find the volume of compressed air that a pipe will carry; given the length and diameter of pipe, initial pressure, and maximum allowable drop in pressure.

Join the initial pressure with the drop, and note the intersection on axis *B*. Join this intersection with the pipe diameter, and note the intersection on axis *A* of a prolongation of this line. A line joining this point on *A* with the pipe length will cut scale No. 2 at the required volume.

3. To find the maximum length of pipe that will carry a given volume of air; given the pipe diameter, initial pressure, and maximum drop.

Join the initial pressure with the drop, and note intersection on axis *B*. Join this point with the pipe diameter, and note

TABLE XVIII (WILLIAM COX)

VALUES OF $\sqrt{\dfrac{p_1 - p_2}{w_1}}$

Final Pressure p_2. Lbs	Losses of Pressure, $p_1 - p_2$. 1 lb.	2 lbs.	3 lbs	4 lbs.	5 lbs	6 lbs	7 lbs	8 lbs.	9 lbs.	10 lbs.
20	2.325	3.241	3.918	4.466	4.930	5.336	5.603	6.014	6.309	6.574
21	2.293	3.198	3.868	4.410	4.870	5.272	5.627	5.946	6.237	6.502
22	2.262	3.157	3.816	4.356	4.812	5.211	5.564	5.878	6.168	6.432
23	2.233	3.117	3.772	4.304	4.756	5.152	5.501	5.814	6.102	6.362
24	2.205	3.079	3.727	4.254	4.702	5.093	5.440	5.752	6.036	6.296
25	2.178	3.042	3.684	4.206	4.649	5.036	5.381	5.688	5.973	6.233
26	2.152	3.007	3.642	4.158	4.597	4.981	5.323	5.630	5.913	6.173
27	2.127	2.973	3.601	4.112	4.548	4.928	5.268	5.572	5.856	6.113
28	2.103	2.939	3.561	4.068	4.499	4.877	5.215	5.518	5.799	6.056
29	2.079	2.907	3.523	4.024	4.452	4.828	5.164	5.466	5.745	5.999
30	2.056	2.876	3.485	3.982	4.408	4.781	5.114	5.414	5.691	5.942
31	2.034	2.844	3.448	3.942	4.365	4.735	5.066	5.364	5.637	5.888
32	2.012	2.815	3.414	3.904	4.323	4.690	5.019	5.312	5.586	5.834
33	1.991	2.786	3.381	3.866	4.282	4.646	4.971	5.264	5.535	5.782
34	1.971	2.759	3.348	3.830	4.242	4.603	4.926	5.216	5.487	5.733
35	1.952	2.733	3.317	3.794	4.202	4.561	4.881	5.170	5.430	5.686
36	1.933	2.707	3.286	3.758	4.164	4.520	4.839	5.126	5.394	5.639
37	1.915	2.682	3.255	3.724	4.126	4.480	4.797	5.084	5.349	5.594
38	1.897	2.656	3.225	3.690	4.090	4.441	4.757	5.042	5.307	5.550
39	1.879	2.632	3.196	3.658	4.054	4.404	4.717	5.002	5.265	5.509
40	1.862	2.608	3.168	3.626	4.020	4.368	4.680	4.962	5.226	5.468
41	1.845	2.585	3.140	3.596	3.987	4.333	4.643	4.924	5.187	5.426
42	1.829	2.563	3.114	3.566	3.956	4.299	4.609	4.888	5.148	5.385
43	1.813	2.542	3.088	3.538	3.924	4.267	4.575	4.852	5.109	5.344
44	1.798	2.521	3.064	3.510	3.895	4.235	4.540	4.814	5.070	5.306
45	1.783	2.501	3.040	3.484	3.866	4.203	4.506	4.778	5.034	5.268
46	1.769	2.481	3.017	3.458	3.837	4.171	4.471	4.744	4.998	5.230
47	1.755	2.462	2.995	3.432	3.808	4.130	4.439	4.710	4.962	5.192
48	1.742	2.444	2.972	3.406	3.779	4.109	4.408	4.676	4.926	5.155
49	1.729	2.426	2.950	3.380	3.752	4.080	4.376	4.642	4.890	5.120
50	1.716	2.407	2.927	3.356	3.725	4.051	4.344	4.608	4.857	5.085
51	1.703	2.389	2.906	3.332	3.698	4.022	4.313	4.578	4.824	5.050
52	1.690	2.372	2.886	3.308	3.671	3.993	4.283	4.546	4.791	5.015
53	1.678	2.355	2.865	3.284	3.645	3.965	4.254	4.516	4.758	4.983
54	1.666	2.338	2.844	3.260	3.620	3.938	4.225	4.484	4.728	4.952
55	1.654	2.321	2.823	3.238	3.596	3.911	4.196	4.456	4.698	4.920
56	1.642	2.304	2.804	3.216	3.571	3.885	4.169	4.428	4.668	4.889
57	1.630	2.289	2.785	3.194	3.547	3.860	4.143	4.400	4.638	4.860
58	1.619	2.273	2.766	3.172	3.524	3.835	4.117	4.372	4.611	4.832
59	1.608	2.258	2.747	3.152	3.502	3.811	4.091	4.346	4.584	4.80[illegible]
60	1.597	2.242	2.730	3.132	3.479	3.787	4.066	4.320	4.557	4.775

TABLE XVIII—*Continued*

VALUES OF $\sqrt{\frac{p_1 - p_2}{w_1}}$

Final Pressure p_2, Lbs	Losses of Pressure, $p_1 - p_2$.									
	1 lb.	2 lbs.	3 lbs	4 lbs	5 lbs.	6 lbs	7 lbs.	8 lbs.	9 lbs	10 lbs.
61	1 586	2 228	2 712	3 112	3 458	3 764	4 042	4 294	4 530	4 747
62	1 576	2 214	2 695	3 092	3 437	3 742	4 019	4 268	4.503	4 718
63	1 566	2 200	2 678	3 074	3 417	3 720	3 995	4 244	4.476	4 693
64	1.556	2.186	2.662	3 056	3.397	3.698	3.971	4.220	4.452	4.668
65	1 546	2 173	2.647	3 038	3 376	3 676	3 948	4 196	4 428	4 642
66	1 537	2 160	2 631	3 020	3 356	3 654	3 926	4 172	4 404	4 617
67	1 528	2 147	2 615	3 002	3 337	3 634	3 905	4 150	4 380	4 592
68	1 519	2 134	2 600	2 984	3.318	3 615	3 884	4 128	4 356	4 566
69	1 510	2 122	2 584	2 968	3 300	3 596	3 863	4 104	4 332	4 541
70	1 501	2 100	2 570	2 952	3 283	3 576	3 842	4 082	4 308	4 516
71	1 492	2 098	2 556	2 936	3 265	3 556	3 820	4 060	4 284	4 494
72	1 484	2 086	2 543	2 920	3 247	3 537	3 799	4 038	4.263	4 471
73	1 476	2 075	2 529	2 904	3 229	3 517	3 778	4 018	4 242	4 449
74	1 468	2 064	2 515	2 888	3 211	3 498	3 759	3 998	4.221	4.427
75	1 460	2 052	2 501	2 872	3 193	3 480	3 741	3 978	4 200	4 405
76	1 452	2 041	2 487	2 856	3 177	3 463	3 723	3 958	4 179	4 383
77	1 444	2 030	2 473	2 842	3 162	3 446	3 704	3 938	4 158	4 361
78	1 436	2 019	2 461	2.828	3 146	3 429	3.686	3 918	4 137	4.339
79	1 428	2 009	2 449	2 814	3 130	3.412	3.667	3 898	4 116	4.317
80	1 421	1 999	2 437	2 800	3 115	3 395	3 648	3 878	4 095	4.294
81	1 414	1 989	2 425	2 786	3 009	3.377	3.630	3 858	4.074	4.272
82	1 407	1 979	2 413	2 772	3 084	3 360	3 611	3.840	4 053	4.253
83	1 400	1 969	2 401	2 758	3 068	3 343	3 593	3 820	4 035	4 234
84	1 393	1 959	2 388	2 744	3 052	3 326	3 575	3 802	4 017	4.215
85	1 386	1 949	2 376	2 730	3 037	3 310	3 559	3 786	3 999	4 196
86	1 379	1 939	2 364	2 716	3 022	3 294	3 543	3 768	3 981	4 177
87	1 372	1 929	2 352	2 702	3 008	3 279	3 527	3 752	3 963	4.158
88	1 365	1 920	2 340	2 690	2 994	3 265	3 511	3 734	3 945	4.139
89	1 358	1 910	2 330	2 678	2 981	3.250	3 495	3 718	3 927	4 120
90	1 351	1 901	2 319	2 666	2.967	3 235	3.479	3 700	3 909	4.101
91	1 345	1 893	2 309	2 654	2 954	3 221	3 463	3 684	3 891	4 082
92	1 339	1.884	2.298	2 642	2 940	3 206	3 447	3 666	3 873	4 064
93	1 333	1.876	2 288	2 630	2 927	3 191	3.432	3 650	3 855	4 048
94	1 327	1 867	2.278	2 618	2 914	3 177	3.416	3 634	3 840	4 032
95	1.321	1 859	2.267	2 606	2 900	3 162	3.401	3 618	3 825	4.016
96	1.315	1 850	2 257	2 594	2 887	3 148	3 387	3 604	3.810	4 000
97	1.309	1 842	2 246	2 582	2 873	3 135	3.373	3.590	3.795	3 984
98	1.303	1 833	2 236	2 570	2 862	3 123	3 360	3 576	3 780	3 969
99	1 297	1.825	2.226	2 560	2 851	3.110	3.347	3.562	3 765	3.953
100	1 291	1 817	2.217	2 550	2 840	3.098	3 334	3.548	3 750	3.937

intersection on axis A. A line joining the point on A with the given cu.ft. of air will cut scale No. 1 at the required length of pipe.

4. To find the pressure at which the air must enter the pipe; given, maximum allowable drop, volume of air, and the diameter and length of pipe.

Join the length of pipe with the volume of air, and note

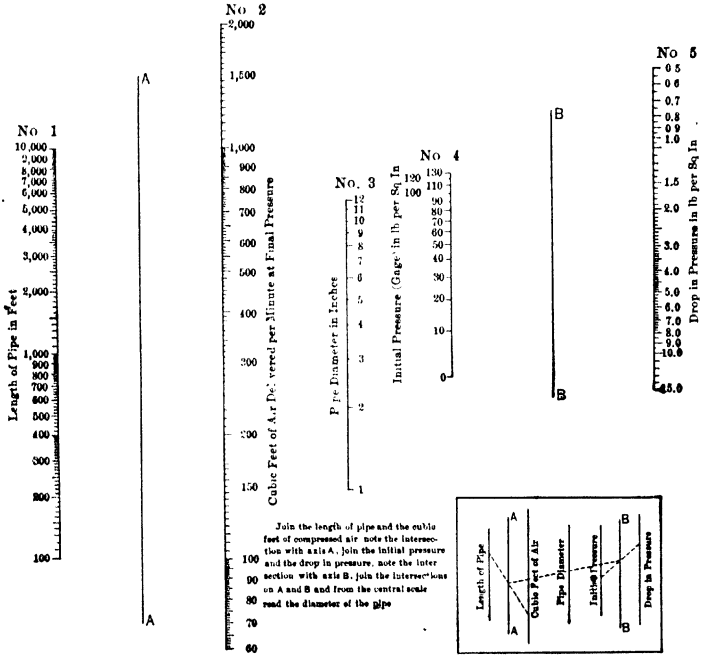


FIG. 121.

intersection on axis A. Join this point with the pipe diameter, and note intersection on axis B. A line joining the point on B with the allowable drop will cut scale No. 4 at the required initial pressure.

5. To find the pressure drop; given, initial pressure, volume of air, and length and diameter of pipe.

Join the pipe length with the volume of air, and note intersection on axis A. Join this point with the pipe diameter, and note intersection on axis B. A line joining the point on B with the initial pressure will cut scale No. 5 at the required pressure drop.

In solving six problems by the nomogram, the mean error was less than 0.5%.

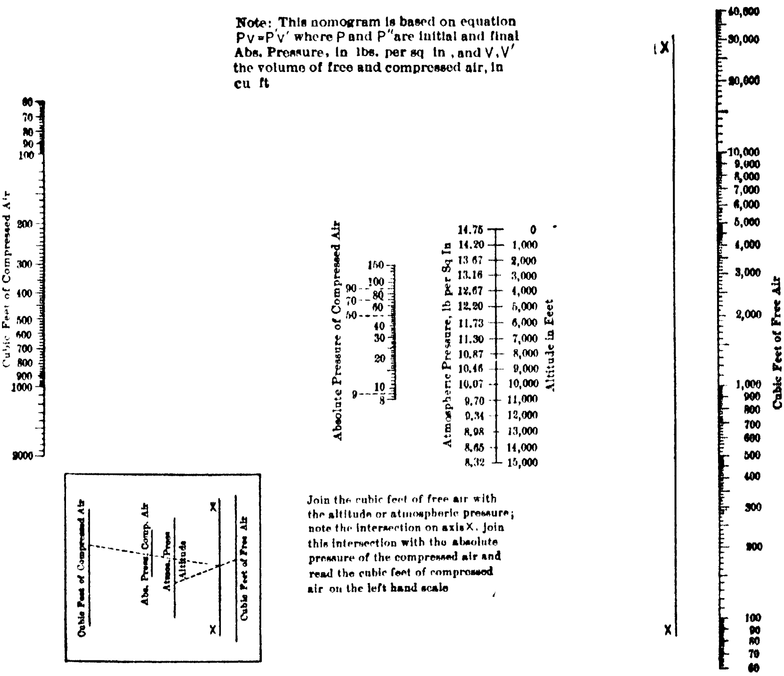


FIG. 122.

NOTE.—The nomogram, Fig. 121, uses volume of *compressed air*, but the compressor capacity and the air consumption of rock-drills are usually expressed in terms of cu.ft. of *free air.* Volumes of free air at different altitudes can be converted into volumes of compressed air at different absolute pressures by means of the nomogram in Fig. 122, which is based on the equa-

tion, PV=P′V′ (Chap. III). The error in using this nomogram is from about −0.15% to +0.5%.

Richards' Formula, best for long pipe lines, is:

$$H = \frac{V^2L}{10{,}000D^5a}$$

in which: D = diameter of pipe, inches; L = length of pipe, ft.; V = volume of compressed air delivered, cu.ft. per min.; H = head or difference of pressure required to overcome friction and maintain the flow, lbs.; a = constant for nominal diameter of pipe.

VALUES OF a FOR DIFFERENT NOMINAL DIAMETERS OF WROUGHT-IRON PIPE

1″	0 350	3″	0 730	5″	0 934
1¼″	0 500*	3½″	0 787	6″	1 000
1½″	0 662*	4″	0 840	8″	1 125
2″	0 565			10″	1 200
2½″	0 650			12″	1 260

* The values of a for 1¼- and 1½-in pipe are not consistent with those for other sizes. See foot-note on p. 223.

By Richards' formula, the calculated losses of pressure are smaller, and, conversely, the volumes of air discharged are larger, under the same conditions, than those obtained from D'Arcy's formula.

The losses of pressure in a table by F. A. Halsey show that the constants used by him differ materially from those given above. Table XIX, containing a series of random examples, shows that in all cases the figures from D'Arcy's formula lie

TABLE XIX

Cu ft Free Air Transmitted at 75 Lbs. (P = 15 Lbs.)	Length of Pipe, Ft	Diameter of Pipe, Ins	Transmission Losses, Lbs.		
			Richards	D'Arcy (Cox)	Halsey.
1,000	1,000	4	3 23	3 71	5 02
1,000	1,000	5	95	1 17	1 63
1,000	1,000	6	35	46	.64
4,000	5,000	8	5 92	8 44	13 05
4,000	5,000	10	1 78	2 81	4 20
4,000	5,000	12	.68	1.06	1.70

between the others; hence it would appear that the results from this formula are sufficiently accurate for ordinary calculations. Within certain limits, the loss of head increases with the square of the velocity. To obtain the best results the velocity of flow in main air pipes should not exceed 20 or 25 ft. per second (Table XX).

TABLE XX*.—(DIAMETER OF PIPE, 12 INCHES)

Velocity of Flow in Ft. per Sec.	Initial Pressure, Lbs.	Final Pressure, Lbs.	Per Cent of Initial Pressure Lost per Mile.
25	100	97.6	2.4
50	100	90.6	9.4
100	100	53.8	46.2

* Unwin. Van Nostrand's Science Series, No. 106, p. 78.

When the initial velocity much exceeds 50 ft. per second the loss becomes large; but by using piping large enough to keep down the velocity the friction loss is almost eliminated. For example, in transmitting 875 cu.ft. of free air per minute at an initial pressure of 60 lbs., through an 8-in. pipe 7,150 ft. long, the average loss including leakage was 2 lbs. The velocity in this case was $8\frac{1}{3}$ ft. per second. A volume of 500 cu.ft. of free air per min., at 75 lbs. gage, can be transmitted through 1,000 ft. of 3-in. pipe with a loss of 4.1 lbs., while if a 5-in. pipe were used the loss would be reduced to .24 lb., the velocities being respectively 28 ft. and 10 ft. per second. In driving the Jeddo mining tunnel, at Ebervale, Luzerne Co., Penna., two $3\frac{1}{4}$-in. drills were used in each heading, with a 6-in. main, the maximum distance of transmission being about 10,800 ft. This pipe was so large in proportion to the volume of air required (about 230 cu.ft. free air per min.) that the velocity was only $3\frac{1}{3}$ ft. per second. A calculation shows a loss of .002 lb., and the gages at each end of the main were found to record practically the same pressure.

Due regard for economy in installation must limit the size of piping, the cost of which in any given case should be considered in relation to the cost of air compression. Diameters of 4–6 ins. for the mains are large enough for 6–10 drills. Up to a length

of 3,000 ft. a 4-in. pipe will carry per min. 480 cu.ft. of free air compressed to 82 lbs., with a loss of 2 lbs. This volume of air will run four 3-in. drills. Under the same conditions a 6-in. pipe, 5,000 ft. long, will carry 1,100 cu.ft. of free air per min., or enough for 10 drills in constant operation. Branch pipes should not be too small. For a length of, say, 100 ft. a 1¼-in. pipe is small enough for one drill, though 1-in. is often used. While it is admissible to increase the velocity in short branches considerably beyond 20 ft. per second, extremes should be avoided. To run a 3-in. drill from a 1-in. pipe 100 ft. long requires a velocity of flow of about 55 ft. per second, causing a loss of 10 lbs. (see Table XXI).

TABLE XXI. (NORWALK IRON WORKS CO.)

Nominal Size of Pipe	1 in		1¼ ins		1½ ins		2 ins		2½ ins	
Length of Pipe in Ft	50	100	100	300	100	300	200	500	250	600
UNIFORM PRESSURE AT THE ENTRANCE OF THE PIPE 80 LBS. GAGE — PRESSURES AT THE POINT OF DELIVERY.	CU.FT. OF FREE AIR DELIVERED AS COMPRESSED AIR AT THE STATED PRESSURES.									
79 8	23 2	16 4	35 2	20 3	63 6	36 7	84 7	53 6	142·0	91 7
79 6	33 1	23 4	49 7	28 7	89 9	51 9	119 6	75 7	200 9	129 6
79 4	40 4	28 6	61 0	35 2	109 1	63 0	146 5	92 7	244 4	157 7
79 2	46 8	33 1	70 3	40 6	127 1	73 4	169 1	107 1	283 2	183 1
79 0	52 3	37 0	78 6	45 4	142 0	82 0	189 1	119 7	317 1	204 6
78 8	57 1	40 4	86 1	49 7	155 4	89 7	207 0	131 0	348 4	224 8
78 6	61 6	43 6	93 0	53 7	168 0	97 0	223 3	141 3	377 0	243 9
78 4	65 9	46 6	99 2	57 3	179 3	103 5	238 7	151 1	399 6	258 4
78 2	70 3	49 7	105 4	60 8	190 5	110 0	252 9	160.1	424 1	273 6
78 0	73 7	52 1	110 8	64 0	200 7	115 9	266 5	168.7	446 7	288 6
77 8	77 2	54 6	116 2	67 1	209 9	121 2	279 2	176.7	469.0	302 6
77 6	80 7	57 1	121 4	70 1	219 1	126 5	291.5	184.5	489 6	315 9
77 4	84 0	59 4	126 3	72.9	228.1	131.7	303.4	192 0	509 3	328 6
77 2	87 1	61 6	131 1	75.7	236 7	136.6	314 4	199.0	528 3	340 8
77 0	90 3	63 7	135 4	78 2	245 2	141 6	325 5	206.0	546 5	352.6
76 8	92 9	65 7	139 8	80 7	252 4	145 7	336 1	212.7	564.2	364 0
76.6	95 6	67.7	143 9	83.1	259 8	150.0	346 2	219.1	581.3	375 0
76.4	98.4	69.6	148 1	85.5	267 6	154 5	356 0	225.3	597 5	385 5
76.2	101 0	71 5	152 1	87 8	274 7	158 7	365 6	231.4	613 8	396.0
76.0	103.8	73 4	156 1	90.1	281 3	162.4	375.6	237 3	629 3	406 0
75.8	106.3	75 2	159 7	92.2	288 4	166.6	383 9	243 0	644 5	415 8
75.6	10[illegible] 7	76.9	163 3	94 3	295 5	170.6	392 8	248 6	659 2	425.3
75.4	111.0	78.5	167.0	96.4	301 7	174.2	401.4	254.0	673 8	434.7
75 2	113.3	80.1	170.4	98 4	307 9	177.8	409 7	259 3	687.8	443.8
75.0	115.5	81.7	173.9	100 4	314.3	181 5	417 9	264.5	701.6	452.7

Compressed-Air Piping is of wrought iron, with sleeve couplings or cast-iron flanges into which the ends of the pipe are expanded or screwed. Sleeve couplings are used for all except large sizes. The smaller sizes, to $1\frac{1}{4}$ in., are butt-welded, while all from $1\frac{1}{2}$ in. up are lap-welded to insure necessary strength. Extra heavy piping may be had for high pressures. Wrought-iron spiral-seam riveted, or spiral-weld steel, tubing, sometimes used, is made in lengths of 20 ft. or less. For convenience of transport in remote regions rolled sheets in short lengths may be had, punched around the edges, ready for riveting, and packed, 4, 6 or more sheets in a bundle.

All joints in mains and branches should be carefully made. The pipe may be tested from time to time by allowing the air at full pressure to remain in the pipe long enough to observe the gage. Leaks should be traced and stopped immediately; they are more expensive than steam leaks, because of the losses already suffered in compressing the air. In putting together screw joints see that no white lead or other cementing material is forced into the pipe; it would make ridges and increase the friction loss. Also, each length should be cleaned of all foreign substances which may have lodged inside. For ready inspection and stoppage of leaks, the pipe should, if buried, be carried in boxes sunk just below the surface of the ground; if underground, it should be supported on brackets along the side of the mine workings. Low points in pipe lines form "pockets" for the accumulation of entrained water, and should be avoided, as they obstruct the passage of the air. In long lines, where a uniform grade is impracticable, provision may be made near the end for blowing out the water at intervals, when the air is to be used for pumps or other stationary engines.

For long lines expansion joints are required, especially when on the surface. Underground they are not often necessary, as the temperature is usually nearly constant, except in shafts or tunnels, where there may be considerable variations of temperature between summer and winter.

As each bend or elbow in a pipe line increases resistance, abrupt changes in direction and sharp curves should be avoided.

For the same diameter of pipe the resistance due to a bend increases as the radius of the curve diminishes. In the absence of exact data the following table is given:

TABLE XXII.—(NORWALK IRON WORKS CO.)

Radius of elbow in terms of diameter of pipe.	5	3	2	$1\frac{1}{2}$	$1\frac{1}{4}$	1	$\frac{3}{4}$	$\frac{1}{2}$
Equivalent length of straight pipe in terms of its diameter	7.85	8.24	9.03	10.36	12.72	17.51	35.09	121.2

These allowances are none too large, since for steam piping the frictional resistance of an ordinary right-angled elbow is considered equivalent to that due to a length of straight pipe equal to 40 times its diameter. But, the usual bends in wrought-iron air piping are not necessarily so sharp as a standard elbow. When many sharp bends are permitted, the resistance may become very great. The matter should have special consideration in the stopes of mines, especially when timbered with square sets; as far as possible, the piping should be carried diagonally through the sets, bending the pipe itself where necessary, instead of using right-angled elbows.

CHAPTER XVII

COMPRESSED AIR ENGINES

COMPRESSED air may be employed as a motive power in an engine in two ways, *viz.*, at full pressure or expansively. By working at full pressure it is understood that the air is admitted to the cylinder throughout practically the entire length of stroke, *i.e.*, without cutoff, and that therefore nearly a cylinderful of air at gage pressure is exhausted at each stroke. In this case the work of the air engine is roughly similar to that done in a non-expansive-working steam engine. Among the machines which use air in this way are rock-drills and simple, direct-acting pumps, without rotary parts.

By the term expansive-working it is meant that the air is admitted to the cylinder during only a part of the stroke, and is then cut off and the stroke completed by the expansive force of the air. For operating in this way some equalizing agent, such as the fly-wheel, is essential, and as a rule a higher initial pressure is employed than when working under full pressure throughout the stroke. It is necessary to distinguish between complete and partial or incomplete expansion. When the air is used with complete expansion the operation in the cylinder is the reverse of adiabatic compression in a compressor, the final pressure being equal to that of the atmosphere. But as air does not undergo condensation, it follows that the lowest terminal pressure in the cylinder must still be sufficiently above atmospheric pressure to produce a proper exhaust, and to overcome the friction of the engine at the end of the stroke. Hence, theoretically complete expansion is impracticable for simple air engines of ordinary design.

Most air engines work with partial or incomplete expansion, the air expanding adiabatically in the latter part of the stroke.

The point of cutoff is such that the terminal cylinder pressure exceeds the back-pressure by an amount sufficient to cause a free exhaust. In the conditions here set forth, no reference is made to the thermal changes incident upon adiabatic expansion in the air cylinder. Although, in principle, compressed air is used like steam, both being elastic fluids, there is an essential difference in the results obtained, due to the reduction in temperature. In expanding behind the piston, a given volume of compressed air at a given pressure will not produce the same amount of power as steam under the same conditions. If two curves be constructed, representing the expansion of equal volumes of air and steam, from the same initial pressure down to pressures below that of the atmosphere, it will be seen that the steam pressure at all points of the stroke is considerably higher than the air pressure; and the expansion curve of the air reaches the atmospheric line sooner than the steam curve.

Fig. 123 shows an ideal card, in which the initial pressure is 75 lbs., and the cutoff is at $\frac{1}{6}$ stroke. The adiabatic expansion curve of the air shows that the pressure is reduced to zero gage pressure when the air has expanded to $3\frac{3}{4}$ times the initial volume, the mean effective pressure being 18.9 lbs. At the end of the stroke the pressure falls to 7 lbs. below atmospheric pressure. The steam curve, on the other hand, does not cut the atmospheric line until the expansion reaches $4\frac{1}{3}$ times the initial volume, and the mean effective pressure is 25.2 lbs. The lower mean pressure of the air is due to the development of cold during its expansion. The operation is the reverse of compression, and the resulting loss of motive power is analogous to the loss of work in the compressor caused by the generation of heat. Just as the heat of compression reacts upon the air while being compressed in the cylinder, and produces a higher tension than that due to the mere reduction in volume; so conversely, when expansion takes place, the air, which is usually at normal atmospheric temperature on entering the cylinder, rapidly gives up its sensible heat, and the cold reacting upon the expanding air reduces its pressure faster than that which is due to the increase in volume alone. Moreover, this behavior of compressed air is independent of the initial

temperature, since the resulting expansion curve would be unaltered. In the case of steam the initial temperature is high, and

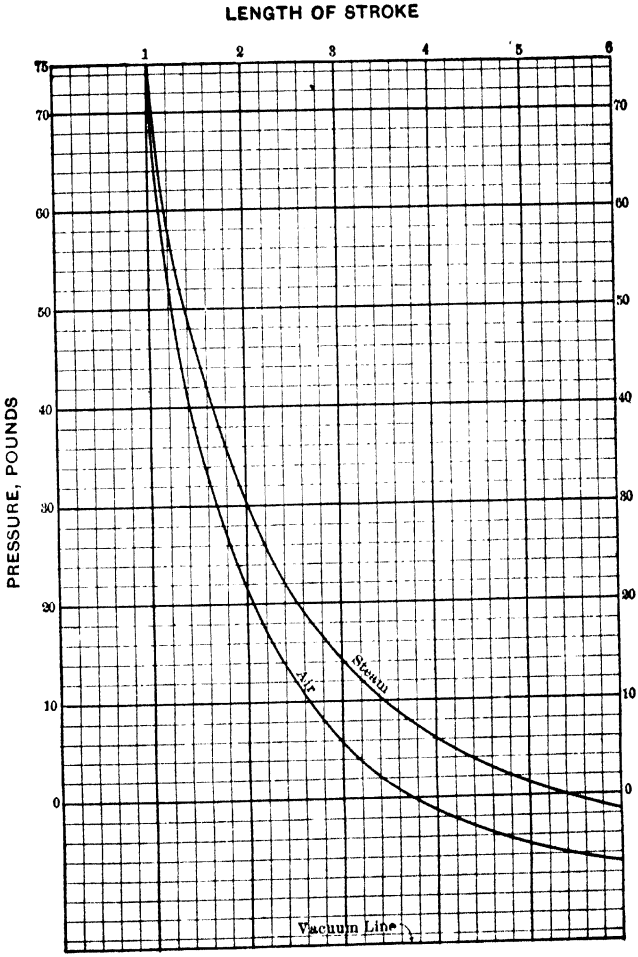


FIG. 123.—Expansion Curves of Steam and Air.

is reduced but little during expansion from ordinary working pressures down to atmospheric pressure.

A similar comparison may be made for other initial pressures and ratios of cutoff. In every case the mean effective pressure is higher for steam than for air. It follows that, to develop the same amount of power in a given cylinder and with the same initial pressure, the cutoff must be later in the stroke with air than with steam.

So low are the temperatures produced by the expansion of air, from ordinary working pressures of 60 or 70 lbs. down to atmospheric pressure, that for a long time the expansive use of compressed air was considered impracticable. In Table XXIII are given the theoretical final temperatures of the exhaust air, in working with complete expansion, and also at full pressure throughout the stroke, for different ratios of initial to final pressure, together with the theoretical efficiencies. The initial temperature is taken as 68° F.*

TABLE XXIII

Ratio of Initial to Final Pressure.	Working with Complete Expansion		Working at Full Pressure.	
	Final Temperature, Deg. F.	Theoretical Efficiency	Final Temperature, Deg. F.	Theoretical Efficiency.
2	−28.2	.855	−8.4	.82
3	−76.0	.806	−34.5	.72
4	−106.6	.782	−45.7	.67
5	−128.2	.768	−54.4	.63
6	−144.4	.758	−59.8	.60
7	−158.8	.751	−63.4	.57
8	−170.8	.746	−66.1	.55
9	−180.6	.742	−68.0	.53
10	−189.2	.739	−69.7	.51

In the table it is shown that by working at full pressure extremely low temperatures of exhaust are avoided; but the efficiency of this method of using compressed air is necessarily much below that obtained from expansive working. It is understood that the temperatures here given are theoretical and are never actually reached in practice. The cold produced is modified by several causes: (1) Some heat is transmitted from the

* M. Mallard, "Etude Théorétique sur les Machines à Air Comprimé," p. 27.

external atmosphere through the cylinder walls; (2) the re-compression of the clearance air at each stroke produces heat in the cylinder, to a degree that increases with the initial pressure and the clearance volume; and, (3) the presence of even a small quantity of moisture in the air tends in some degree to raise the cylinder temperature.

A few brief notes will here be given concerning the elements of the operation of compressed-air engines, that may be considered more or less applicable for ordinary service, *viz.*, working at full pressure, with partial expansion, or with complete expansion. Isothermal expansion may be neglected, since it involves the application of a sufficient degree of external heat to the air while doing its work in the cylinder to produce a terminal temperature equal to the initial temperature.

1. **Working at Full Pressure.** This mode of using compressed air is common for engines like pumps, operating under a constant resistance and not provided with fly-wheels:

Let P' = the absolute initial pressure of the air;

V' = the initial volume of air, at the pressure P', or K times the volume of 1 lb. of air used per unit of time;

T' = the absolute initial temperature of the compressed air;

T = the absolute final temperature of the air at exhaust, on expanding to atmospheric pressure;

P = pressure of the air at exhaust;

W = foot-pounds of work done.

From the theory of compressed air:

$R = J(C_p - C_v) = 778(0.2375 - 0.1689) = 53.37$, where J is Joule's heat unit, and C_p and C_v are the specific heats of air at constant pressure and constant volume.

As no work is done by the expansive force of the air originally produced by compression, W equals the volume of air used, V', multiplied by the difference between P' and P, or:

$$W = V'(P' - P)$$

Substituting for V' its value, $\frac{KRT'}{P'}$:

$$W = \frac{KRT'}{P'}(P' - P) = 53.37KT'\left(1 - \frac{P}{P'}\right)$$

2. **Working with Partial Expansion.** The advantages of using compressed air in this way may be obtained from engines possessing fly-wheels, provided that the cutoff be not too early in the stroke to avoid excessive reduction of cylinder temperature, or else that the air be reheated before entering the cylinder.

In this case the values of P', V', and T' are as above. From the point of cutoff the air expands adiabatically down to a terminal pressure of P'' and volume V'', the final temperature in the cylinder falling to T''. On exhausting, the pressure, volume, and temperature become P, V, and T. The work done is composed of three parts, *viz.*,

W' = work between the point of admission and the point of cutoff $= P'V'$;

W'' = work performed by expansion of the volume V' from the point of cutoff to the end of the stroke $= 778\ KC_v (T'-T'')$;

W''' = negative work due to back-pressure $= -PV''$.

Taking the algebraic sum of these three quantities:

$$W = P'V' + 778KC_v(T'-T'') - PV''$$

But, as under (1): $V' = \frac{KRT'}{P'}$ and $V'' = \frac{KRT''}{P''}$

Substituting these values of V' and V'', and for R and C_v, their numerical values of 53.37 and 0.1689:

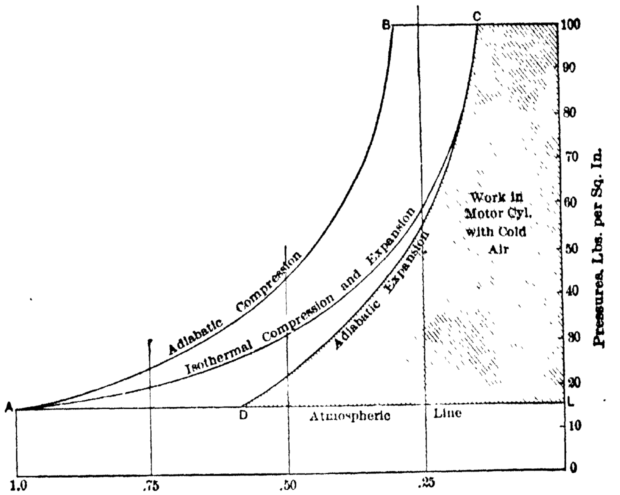
$$W = K\left[53.37T' + 131.4(T'-T'') - 53.37T'\left(\frac{P}{P''}\right)\right]$$

$$= 53.37K\left[T' + 2.46(T'-T'') - T'\frac{P}{P''}\right]$$

3. **Working with Complete Expansion.** In the theoretical card, Fig. 124, are shown the relations of the compression and expansion lines, the shaded portion representing the useful work done by the complete expansion of cold air in a motor cylinder. When the expansion is adiabatic, the same relations exist between

pressures, volumes, and temperatures as were set forth in the discussion of adiabatic compression, *viz*:

$$\frac{P'}{P}=\left(\frac{V}{V'}\right)^{n=1.406}=\left(\frac{T'}{T}\right)^{\frac{n-1}{n}=0.29}$$



Vols. in Cu. Ft.
FIG. 124

The theoretical work done by complete adiabatic expansion may be expressed by a formula like that employed for compression, but with an inversion of certain of the quantities, thus:

$$W=\frac{n}{n-1}PV\left[1-\left(\frac{P}{P'}\right)^{\frac{n-1}{n}}\right],$$

in which W = theoretical foot-pounds of work done by the expansion to atmospheric pressure of 1 lb. (13.1 cu.ft.) of free air. Substituting the values of the constants, and for working at sea-level:

$$W=3.463\times144\times14.7\times13.1\times\left[1-\left(\frac{14.7}{P'}\right)^{0.29}\right]$$

$$=96,029\left[1-\left(\frac{14.7}{P'}\right)^{0.29}\right]$$

For example, if P′ be 40 lbs. gage pressure:

$$W = 96{,}029\left[1 - \left(\frac{14.7}{54.7}\right)^{0\ 29}\right] = 30{,}440 \text{ ft. lbs., or } 2{,}323 \text{ ft.-lbs. per cu.}$$

ft. of free air.

Actual Work Done. In the above expressions no account is taken of the friction of moving parts of the motor engine, or loss of work caused by leakage. In determining the actual work, the general case will be where a cutoff is employed. The relations between initial and terminal pressures and temperatures, for different ratios of expansion in a motor-engine cylinder, are shown in Table XXIV,* the points of cutoff, in tenths of the cylinder stroke, being given in the first column.

TABLE XXIV

THEORETICAL RATIOS OF PRESSURES AND TEMPERATURES DUE TO THE EXPANSION OF COMPRESSED AIR IN A MOTOR CYLINDER

Cutoff	Ratio of Expansion = 1 ÷ Cutoff.	Ratio of Mean to Total Absolute Pressure for Entire Stroke.	Ratio of Mean to Total Absolute Pressure, During Expansion Only	Ratio of Initial to Final Temperature	Ratio of Initial to Final Absolute Temperature Due to Expansion Only	Ratio of Initial to Final Absolute Pressure for Ratio of Expansion.
0 10	10 00	0 249	0 166	0 391	0 513	0 039
15	6 67	348	233	460	578	069
20	5 00	436	295	518	627	104
25	4 00	515	353	568	669	142
30	3 33	585	408	612	705	184
35	2 86	647	460	652	737	.228
.40	2 50	706	510	.688	767	.275
.45	2 22	757	558	722	794	325
50	2 00	.802	604	754	818	.378
55	1.81	842	649	784	841	.433
.60	1 67	877	692	812	862	.487
.65	1 54	907	734	839	882	545
.70	1 43	932	774	865	902	605
.75	1 33	954	814	889	920	667

* This table, as well as Table XXV, is taken in part from those used by G. D. Hiscox, in "Compressed Air, its Production, Uses and Application," 1901, p. 202.

The quantities in Table XXIV must be further corrected for piston clearance and the lost volume represented by the air ports and passages of the cylinder, because part of the air expands into these clearance spaces. Therefore, the actual effect of the cutoff, in any given case, is found by dividing the sum of the cutoff plus clearance, by the cylinder volume plus clearance. For example, if the stroke be 10, with a cutoff of $\frac{4}{10}$, and clearance of 6%, the actual volume of the cylinder, including clearance, will be: $(10\times.06)+10=10.6$. Then the sum of the cutoff plus the clearance is $4+.6=4.6$, and the working cutoff becomes $4.6\div10.6=0.434$. In this way Table XXV has been constructed, for use in connection with Table XXIV. It shows the actual cutoff corresponding to the different nominal points of cutoff, for the percentages of piston clearance named at the top of the columns.

TABLE XXV

ACTUAL CUT-OFF DUE TO CLEARANCE, FOR THE NOMINAL CUT-OFFS IN COLUMN 1

Nominal Cutoff	Percentage of Clearance						
	.03	.04	.05	.06	.07	.08	.10
10	0 126	0 135	0 143	0 151	0 159	0 167	0 182
15	175	184	191	198	206	213	227
.20	223	231	238	245	252	259	273
25	272	279	286	293	299	305	318
.30	320	327	333	340	346	352	364
.35	368	376	380	387	392	398	409
40	417	423	429	434	439	444	455
45	465	471	477	481	486	490	500
.50	514	519	524	528	533	537	546
.55	564	568	571	576	.580	585	591
60	612	615	619	623	626	630	637
.65	660	664	667	670	673	676	682
70	709	711	714	717	720	.722	.727
.75	758	760	762	764	766	.768	.772

The theoretical terminal cylinder pressure resulting from adiabatic expansion may be expressed by:

$\frac{P'}{C^{1.406}} - P$, in which C = ratio of expansion $= \frac{1}{\text{point of cut-off}}$ (see column 2, Table XXIV).

For example, for a cutoff at $\frac{4}{10}$ stroke and 65 lbs. gage pressure, the terminal pressure (above atmospheric pressure) will be:

$$\frac{65+14.7}{2.5^{1.406}} - 14.7 = 7.2 \text{ lbs.}$$

The volume corresponding to the nominal cutoff is increased by the clearance, and adds to the mean pressure. Thus, in the above example, assuming the clearance to be 6%, the actual cutoff (Table XXV) is increased from 0.4 to 0.434, of which the ratio C is $\frac{1}{.434} = 2.3$. From Table XXIV, column 7, the ratio of initial to terminal pressure, corresponding to the actual cut-off of 0.434, is (by interpolation) .31; whence: $(79.7 \times 0.31) - 14.7 = 10$ lbs. terminal pressure.

Cylinder Volume Required for a Given Power. The work per stroke is found by dividing the foot-pounds of work to be done per minute by twice the number of revolutions of the engine (which would be determined for any given size of engine by the ordinary empiric rules of practice). This is substituted, with the initial and final pressures, in the formula for working with full pressure, partial or complete expansion, as the case may be, which is then solved for the initial volume, V′, of compressed air used per stroke. To the theoretical cylinder volume thus found, the allowance for piston clearance is added, according to the type of engine. The proper proportion between stroke and diameter of cylinder is finally determined.

The volumes of free air per minute, required for an air engine, per indicated horse-power and for different ratios of cutoff, are shown in Table XXVI, by F. C. Weber.* The figures given in this table do not include the volume corresponding to piston clearance, which may be found as already shown.

* *Compressed Air*, Oct., 1896, p. 117.

TABLE XXVI

CUBIC FEET OF FREE AIR PER MINUTE USED IN MOTOR ENGINE, PER I.H.P.

Point of Cutoff	Gage Pressures, Lbs.									
	30	40	50	60	70	80	90	100	110	125
1	23.3	21.3	20.2	19.4	18.8	18.42	18.10	17.8	17.62	17.40
$\frac{3}{4}$	18.7	17.1	16.1	15.47	15.0	14.6	14.35	14.15	13.98	13.78
$\frac{5}{8}$	17.85	16.2	15.2	14.5	14.2	13.75	13.47	13.28	13.08	12.90
$\frac{1}{2}$	16.4	14.5	13.5	12.8	12.3	11.93	11.7	11.48	11.30	11.10
$\frac{1}{3}$	17.5	15.2	12.9	11.85	11.26	10.8	10.5	10.21	10.02	9.78
$\frac{1}{4}$	20.6	15.6	13.4	13.3	11.40	10.72	10.31	10.0	9.75	9.42

In this table the air is supposed to be used without reheating, and at an initial temperature of 60° F. Reheating will reduce the volume of air proportionally to the ratio $\frac{T_2}{T_3}$, where $T_2 = 459° + 60° = 519°$ F., or absolute temperature; and $T_3 = 459°$ plus the temperature of the reheated air on entering the motor cylinder. Thus, if the air be reheated to 200° F., the above ratio becomes $\frac{519°}{659°} = 0.787$, by which decimal the volume of air as found in the table must be multiplied.

So far as mine service is concerned, it has been customary to consider compressed air almost exclusively as an agent for the operation of rock-drills, and in view of its preponderating application to this use its adaptability under proper conditions to the driving of other machines and engines is sometimes overlooked. Of late years, however, with improved methods of compression and reheating, attention has been given to employing compressed air for a greater variety of service; not only underground, but for certain portions of the surface plant of mines as well. Aside from cases where the disposal of exhaust steam would be troublesome, the question is largely one of comparative loss in transmission and the power cost of the air.

Although not strictly in place in this chapter, reference may be made to what has been called the "two-pipe system" or

"high-range compressed-air transmission," introduced some years ago by Charles Cummings.*

The machine or engine using the air makes in effect a closed circuit with the compressor. After the air has done its work in the motor cylinder, it is returned to the compressor at the pressure of the exhaust, through a second line of piping. The return pipe connects with a closed chamber at the compressor, in which the inlet valves are placed, thus enabling the compressor to begin its stroke with the cylinder filled under a considerable initial pressure. Then, after raising the pressure to the original point, the compressor delivers the air into the main, to be used again by the air engine. The actual working pressure of the air engine is, therefore, the difference between the pressures in the delivery and return pipes. Barring leakage, the same air is thus used over and over, the intention being that the compressor shall put back into the air kept in circulation the power expended in the motor-engine cylinder.

Though the compressor itself is not materially different from the ordinary forms, the two-pipe system requires a rather complicated arrangement of piping and valves for charging the apparatus with air at the working pressure adopted, and for governing the speed and output according to the rate of consumption of air.† The advantages of the system are: a higher efficiency than is obtained from moderate-size compressors of the usual types, and less trouble from freezing at the motor engine by reason of the relative dryness of the air due to its higher tension. The efficiency increases with the pressure employed. In using compressed air without reheating the two-pipe system is superior in principle to the ordinary mode of operating compressed-air plant. But because of the greater first cost its advantages disappear when reheating can be adopted, and the single-pipe system is then found to be preferable.

The two-pipe system is best suited for machines working

* Patent No 456,941 was issued to Mr. Cummings in 1891.

† A detailed illustrated description is given by Frank Richards in *American Machinist*, April 28, 1898, p. 23. See also *Compressed Air Magazine*, Oct., 1907, p 4599.

at full pressure throughout the stroke, such as machine drills or simple, direct-acting pumps. When the motor works expansively the pulsations become objectionable, as a regular flow of air is not maintained in the return pipe. Under these conditions the inertia and friction of high-pressure air in long pipe lines becomes noticeable and disadvantageous.

As the length of air pipe required for this system is doubled, not only may the first cost of the pipe go far toward offsetting the greater efficiency but, with at least twice as many joints in the pipe lines, the chances of loss from leakage are increased. And if very high pressures be used (pressures of several hundred pounds have been proposed), not only must the piping itself be heavier and more expensive, but the proportionate power loss from leakage is greater. For moderate distances, however, and when working at full pressure under the proper conditions, the foregoing disadvantages may be more than counterbalanced by the superior efficiency of the system. Though not yet in general use, the two-pipe system is said to have given satisfaction at several mines in New Mexico, Colorado, and California,* and in 1905 was proposed for use in the Johannesburg gold district. Some prominence is here given to the system because of its novel features and the probability that it may be found useful, if its disadvantages can be overcome. In a paper by H. C. Behr, published in 1905 in the *Transactions of the Mechanical Engineers' Association of the Witwatersrand*, the Cummings system is treated at length, with a discussion of its advantages for air-driven pumps.

Compressed-Air Hoists. In a few mines, compressed-air engines of considerable size are used in deep shafts, where the hoisting is in two stages. The usual case is where an inclined shaft is sunk in the vein, or in the footwall rock, to a point of intersection with a vertical shaft. The compressed-air hoist at the head of the incline delivers ore into pockets at the foot of the vertical shaft, from which the main steam or electric hoist raises it to the surface. This plan is followed at several mines on the Witwatersrand, South Africa, as at one of the

* A. E. Chodzko, *Modern Machinery* (Chicago), Jan., 1899, p. 11.

Simmer Deep Shafts. For small-scale work standard geared hoists are used, usually without trouble from freezing of moisture, because in intermittent work the cylinders have time to regain normal temperature.

For heavy work the cylinders should be designed for expansive use of the compressed air. The clearance volume is thus reduced, and larger admission and exhaust ports are required, because at the same pressure the density of air exceeds that of steam. Loss in efficiency due to incomplete expansion can be reduced by compounding the cylinders, and, if possible, reheating the air. With a cutoff in the high-pressure cylinder at 0.9 stroke (the minimum practicable starting cutoff), and reheating between the cylinders to the initial temperature, the loss is about one-half that of a simple cylinder, or a saving of 25% of the energy in the air entering the high-pressure cylinder. At the Miami mine, Ariz., the reheating temperature is 350°–375° F.; at the Anaconda mine, 250°–350° F. The volume of air required and the results with different cutoffs are discussed on previous pages (see also Table XXVII).

TABLE XXVII

VOLUME OF FREE AIR (60 LBS. GAGE) FOR DUPLEX HOISTS

Diameter of Cylinder, Ins.	Strokes, Ins.	Revs per Min.	Normal H P	Actual H.P.	Weight Lifted, Single Rope, Lbs	Free Air per Min., Cu.ft.
5.	6	200	6	11 8	1,000	300
5	8	160	8	12 6	1,650	320
6 25	8	160	12	19 8	2,500	500
7	10	125	20	24 2	3,500	604
8 25	10	125	30	33 6	6,000	680
8.50	12	110	40	37 8	8,000	952
10.	12	110	50	52 4	10,000	1,320
12.25	15	100	75	89 2	....	2,250
14.	18	90	100	125.	....	3,174

Small underground compressed-air hoists, portable or semi-portable, are furnished by several makers for sinking shafts from level to level, sinking winzes, raising timbers into position

in stopes, temporary haulage of cars, and other light work. They are mounted on a drill column or bar, or can be bolted to a timber. Following are examples:

The Ingersoll-Rand "Little Tugger" (Fig. 125) is made in two sizes: "I-H," for 300 ft. of $\frac{5}{16}$-in. or 500 ft. of $\frac{1}{4}$-in. wire rope, to raise a 1,000-lb. load, and "H-H," for 200 ft. of $\frac{3}{8}$-in. manila rope, to raise 600 lbs. It will run on compressed air or steam. The engine is of the square piston type, giving 4 impulses per min., so that there is no dead center. With air at

FIG. 125—Ingersoll-Rand "Little Tugger" Hoist, Type "I-H."

80 lbs., a load of 1,000 lbs. can be raised at a speed of 85 ft. per min. (=about 2.5 H.P.).

The Leadville "column-hoist" (Mine and Smelter Supply Co.) has a 2-H.P. engine. Other designs are the Holman, made in sizes of 2, 4 and 6 H.P., and the portable hoist, for column mounting, of the Chicago Pneumatic Tool Co., 2 H.P., lifting capacity, 650 lbs., at 90 ft. per min.; 200 ft. of $\frac{5}{16}$-in. rope; weight, without rope, 300 lbs.

Anaconda Copper Co's Compressed-Air Hoisting Plant.*

* For further details, see papers by. B. V. Nordberg, *Trans.* A.I.M.E., Vol. XLVI, p. 826; and *Eng. & Min. Jour.*, May 18 and 25, 1912. See also: K. A. Pauly, *Trans.* A.I.M.E., Vol. XLII, p. 533; R. R. Seeber, *Eng. & Min. Jour.*, Aug. 3, 1912, p. 197, and T. T. Read, *Min. & Sci. Press*, Nov. 2, 1912, p. 554.

From 1911 to 1913, a number of the main Anaconda hoists were changed to compressed-air drive by the Nordberg Mfg. Co. Most of the shafts range in depth from about 2,000 to 2,500 ft. The great variation in power required in deep hoisting, and the high peak loads during the period of acceleration, suggested the use of compressed air, in connection with an extensive storage system. At one double-compartment shaft, where large quantities of waste rock for filling were being lowered from the surface, the indicated horsepower ranged from $-1,600$ to $+2,300$. An elaborate series of tests on the former steam-hoisting plants led to the following plan:

A. An electrically operated compressor plant, with a capacity of 7,650 cu.ft. of free air per min. compressed to 90 lbs., the electric current being generated by water power, 150 miles from Butte.

B. Air storage of large capacity. This is connected with the air piping for the rock-drills, so that, in periods when little or no air is used by the hoists, all the air could be turned into the drill mains. To take care of the peak loads, each hoist has several air receivers, aggregating about 8,000 cu.ft. capacity. Maximum air consumption of the largest hoists is 60 cu.ft. of compressed air per second, for a period of 5 seconds; the heaviest unbalanced load requires an average of 40 cu.ft. of compressed air per second for about 1 min., or a total of 2,400 cu.ft. The pipe lines are proportioned for a pressure drop of 5 lbs. At times when several large hoists are running simultaneously there is a total draft of 2,600 cu.ft. compressed air per min., or 22,130 cu.ft. free air (atmospheric pressure at Butte being 12 lbs.). The combined maximum consumption of the entire 27 hoists would be 37,200 cu.ft. free air per min. (average, 27,660), the highest peak representing a volume of 44,000 cu.ft. As this measures the minimum required storage capacity to equalize the load on the system, to provide for it, and allowing 10% pressure drop, the total receiver capacity would be 440,000 cu.ft., which is prohibitively large.

A hydrostatic storage plant was therefore installed, consisting of a number of connected air receivers of 66,000 cu.ft. total

capacity. About 210 ft. higher than the receivers (corresponding to a pressure of 91.15 lbs. per sq. in.) is an open water tank, 100 ft. diameter, with a pipe leading from its bottom to the receivers. When 66,000 cu.ft. of compressed air have been delivered into the receivers, the water in them is displaced and forced into the upper tank. Hence, the hoists are supplied with air at constant pressure of 91.15 lbs., less the pressure drop due to pipe friction.

C. Air reheaters are an important feature of the plant, the heating medium being steam at 200 lbs. pressure; reheating temperature, 300° F.* Coal used for reheating is ⅓ lb. per H.P. hour.

D. The steam hoisting engines, ranging in sizes from 28 ins. by 48 ins. to 34 ins. by 72 ins., were fitted with new cylinders, with special valve gearing.†

The advantages of the system are: (*a*) the existing hoists were retained, with comparatively few changes; (*b*) the engines may be run by steam if necessary; (*c*) the air storage is sufficient to operate each hoist for about 20 min., in case of stoppage of the electric power, due to thunder storms, etc.; (*d*) hoisting capacity can be increased by increasing speed, and the same engine will hoist from a greater depth by increasing the compressor capacity; (*e*) part of the energy liberated during the period of retardation, or when waste rock is being lowered, is returned to the power system; (*f*) peak loads on the power plant are eliminated.

Approximate efficiency of the main hoists:

Electric motor efficiency	95%
Total efficiency of the compressors	74
Efficiency of hoisting engines	50
Total efficiency of plant, without reheating	35 4
Total efficiency of plant, with reheating	53

Other Compressed-Air Hoists have been installed at several mines.

* For details see *Trans*. A I.M E., Vol XLVI, p. 857.

† *Ibid*., p 860. See also Peele's Mining Engineers' Handbook, Sec. 12, Art. 1c.

I. Miami Copper Co., Ariz. A 20 in. by 48 in. Nordberg geared hoist, with 10-ft. drums. Air at 80 lbs. is reheated to 370° F.; consumption, 2,275 cu.ft. free air per min., for depth of 675 ft. Rope speed, 750 ft. Capacity, 287 tons per hour. A rough test showed the net shaft H.P. to be about 60% of the I.H.P. of compressor Cost, f.o.b. Milwaukee, $22,500.

II. Franklin Junior mine, Mich., has a combined air and steam hoist.* The 10-ton empty skip is lowered unbalanced, and compresses air back into 3 steam and air storage receivers, each 10 ft. diam. by 32 ft. long. Air cylinders, 36 ins. diam., are tandem to 46-in. steam cylinders; stroke, 72 ins.; Corliss valve gear. Drum, 15 ft. diam., holds 5,130 ft. of $1\frac{5}{8}$-in. rope. Two 200-H.P. boilers are used. When hoisting, the ends of the air cylinders are by-passed, and no air is compressed; when lowering, the steam-cylinder exhaust valves are open, and the air compressed in the air cylinders is discharged into the receivers. A reducing valve keeps the receiver pressure at 75 lbs. The skip is lowered with the receivers full of steam at 75 lbs.; the compressed air raises the pressure to 95 lbs., which is then available to start hoisting. When the pressure falls below 75 lbs., steam is admitted to complete the hoist.

III. Copper Queen mine, Ariz., has 3-direct-acting and 2 geared hoists operated by compressed air. The larger direct-acting hoists use 1,400–1,600 cu.ft. free air per shaft H.P.

* R. H. Corbett, *Eng. & Min. Jour*, Sept. 21, 1912, p. 553; see also *Power*, Nov. 7, 1916.

CHAPTER XVIII

FREEZING OF MOISTURE DEPOSITED FROM COMPRESSED AIR

REFERENCE has been made in a former chapter to the trouble sometimes caused by the congelation of the moisture carried in compressed air when deposited in the transmission pipes or in the ports and exhaust passages of the machine using the air. The presence of moisture in compressed air must be accepted as an unavoidable condition. Existing in the atmosphere at all times in greater or less quantity, when air is compressed the moisture is carried with it. A part of the water is deposited in the air receiver, but a considerable quantity still remains and will be brought into evidence when the proper conditions occur.

The capacity of air for moisture depends primarily upon its temperature. Under ordinary atmospheric conditions 1,000 cu.ft. of air contain about 1 lb. of water. When its volume is reduced in the compressor cylinder, the increase of heat which takes place augments its moisture-carrying capacity. Any subsequent decrease in temperature reduces this capacity, and if the air be saturated the excess of moisture is deposited. Volume for volume, the capacity of air for moisture is independent of its pressure or density. That is, at the same temperature, a cubic foot of air at atmospheric pressure will hold in suspension the same weight of water as a cubic foot at 100 lbs. pressure. But this must not be misunderstood. If a certain volume of saturated atmospheric air be compressed isothermally, say to $\frac{1}{10}$ of its original volume, its water capacity is also reduced to $\frac{1}{10}$, and $\frac{9}{10}$ of the water originally present in the air is deposited. Therefore, while the capacity for carrying moisture of a given volume of air varies with the temperature, it must change also with any increase or decrease of pressure which changes its volume.

Causes of Freezing. Certain conditions are required to cause freezing of compressed air: deposited moisture must be present, and it must be subjected to a temperature below the freezing-point. So long as the temperature does not fall low enough, the presence of moisture can do no harm. Although one of the recognized functions of the air receiver is to permit the deposition of water before the air passes into the pipes, still, unless the receiver be extremely large, the air leaves it warm—usually even quite hot—and therefore carries with it considerable moisture. In the case of wet compressors, unless liberal sprays are used to attain effective cooling, the air is apt to contain more moisture than that from dry compressors. A well-designed injection compressor, however, not too small for its work and therefore running at a moderate speed, will deliver cool air which will not give trouble from freezing. The air having attained nearly normal temperature before entering the pipe-line, its moisture-carrying capacity undergoes but little further reduction while passing through the pipe, and only a small amount of additional deposition takes place. With dry compression the percentage of humidity of the intake air, and the temperature at discharge, determine the quantity of water carried out of the cylinder. The humidity, in turn, varies with the weather. Changes in the weather may quickly be followed by variations in the quantity of moisture deposited in the receiver and pipe-line. When the air is finally expanded in doing its work in the air engine, intense cold is produced as the pressure falls, and the latent heat of compression is absorbed. It is here that the moisture carried with the air into the pipes makes its appearance as frost and causes trouble. Watery vapor itself, depositing a light, snow-like frost, does not tend to clog the air passages and ports as much as entrained water in a finely divided state, which will gradually form accumulations of solid ice and choke the exhaust wholly or in part.

Prevention of Freezing. The difficulties which may arise from the conditions just outlined are apt to be exaggerated. That freezing not infrequently occurs is true, but with a properly designed and arranged plant it may easily be avoided. Two

things require attention: *first*, the air should be caused to drop its moisture as completely as possible before entering the main; *second*, provision should be made for draining off what deposited moisture remains in the pipe-line, before the air passes to the machine in which it is to be used. Although this is a simple matter, the means for accomplishing it are often neglected. Considerable quantities of water may collect in low places in the pipe-line and, if not blown out at intervals, will be carried into the ports, cylinders, and exhaust passage of the air machine and there freeze.

Granting that the air leaves the receiver near the compressor practically saturated and still warm, it is evident that a great improvement in working conditions may be realized by introducing a second receiver as close as possible to the machines using the air. In mining the second receiver is, of course, placed underground.* Before reaching it, the temperature of the air will have become normal, and the entrained moisture from the pipe-line may readily be trapped and drawn off. It may be remarked that automatic water-traps are preferable to valves or cocks for getting rid of the water. As a rule, when the compressed air is to be used expansively, a special aftercooler should be introduced, placed as close as possible to the compressor. In any case, the receiver should be of ample size to insure the deposition of the moisture. The advantages of reheating the air before use will be taken up later.

Influence upon Freezing of High Pressures in Transmission. The statements made in the first part of this chapter suggest an important consideration, *viz.*, in transmitting power by air at a high pressure there is less liability to trouble from freezing than when low pressures are employed, provided that the length of pipe-line is sufficient to allow the air to be completely cooled and drained of its water while still under high pressure. At a low pressure a greater volume of air is required to furnish a given amount of power than when at a high pressure. More moisture must, therefore, be dealt with, and at the low pressure it cannot be so thoroughly separated before the air is used. Suppose the

* See Chapter XI.

transmission to be at a high pressure, and through a pipe long enough to allow the air to reach normal temperature. If the deposited moisture be drained away while the air is at its maximum pressure; then, if the air be subsequently expanded down to a lower pressure suitable for working (with a corresponding increase of volume) and allowed to regain its normal temperature, the percentage of moisture will be reduced, so that the air may be relatively very dry. When finally used in the air engine there will not be enough moisture present to cause troublesome freezing.

Deposition of Moisture by Reducing Pressure. Still another mode of minimizing trouble from freezing is to reduce the pressure of the air before it enters the cylinder of the air engine. The means by which this is accomplished and the results obtained may be illustrated by an example.

At the Drummond Colliery, Nova Scotia, for running an underground pump by compressed air two receivers are used, one near the pump, and another 300 ft. farther back on the pipe-line. The air pressure in the main from the surface is 85 lbs., and as the proportions of the cylinders of this particular pump are such that so high a pressure was unnecessary a reducing valve was put in the pipe just before reaching the first receiver. By this valve the air is wire-drawn to reduce the pressure to 45 bs., which results in a deposition of nearly one-half the entrained water, in addition to that already deposited in the pipes. It is found that more moisture collects in the first than in the second receiver, and by this device the serious difficulty previously encountered from freezing at the pump has been entirely overcome.* The temperature lost by the reduction of pressure to 45 lbs. is regained before the air reaches the pump.

Protection of Surface Piping. What precedes refers only to the freezing produced by internal reduction of temperature, acting on the moisture carried in the air. In using compressed

* This information has been kindly furnished by Charles Fergie, superintendent of the Drummond Colliery. See also Mr. Fergie's article on the subject, in *Transactions Canadian Mining Inst.*, 1896, of which an abstract was published in the *Colliery Guardian*, October 30, 1896, p. 821.

air, even for mining purposes, it often becomes necessary to carry lines of air pipe considerable distances on the surface. To prevent condensation and freezing of the moisture in winter by external cold, all surface piping must be protected. If exposed to temperatures below the freezing-point, the inside of the pipe will become coated with ice and its effective cross-section reduced. A serious diminution of area may thus be caused at low points in the pipe-line, where water tends to collect; or the pipe may even be frozen solid in such places by the gradual accumulation of ice. Underground the temperature is rarely, if ever, low enough to render any protection necessary, except in cold, down-cast shafts, or in tunnels in which there is a strong inward draught.

Some time ago, at one of the Butte copper mines, a simple and inexpensive device was employed to prevent the freezing of moisture in a long line of surface piping. The air main of a large compressor plant was carried on the surface some hundreds of feet before reaching the shaft. During the winter months it was at times difficult to get sufficient air pressure in the mine because of the partial choking up of the pipe. As the volume of compressed air was too large to be dealt with by the ordinary receiver, a series of old tubular boilers were placed close to the compressor house. The hot air, at 80 lbs. gage pressure, in passing through these boilers, from one to another, was cooled down practically to atmospheric temperature and as a consequence a large part of its moisture was deposited. It was found that discarded tubular boilers, when strong enough, were well suited to this purpose, because of the large surface presented to the cold outside air; especially when they are set horizontally, so that there is a free circulation of air through the tubes. A blower might be used for the same purpose in a warm climate, or the boilers submerged in cold water. This effectual remedy is worthy of adoption where the conditions are similar.

CHAPTER XIX

REHEATING COMPRESSED AIR

After the warm compressed air enters the transmission line its temperature is quickly reduced to that of the surrounding atmosphere. The facility with which the heat of compression is given up suggests the gain that may be effected by reheating the air when it reaches the place where it is to be utilized. By reheating an added volume of air is obtained at a lower power cost than if it were produced by a compressor. This is shown by comparing the heat units required to produce a given volume of air at a given pressure in a compressor cylinder with the heat units required to accomplish the same result by expanding the air by the direct application of heat.

Appliances for, and Results of Reheating. The most important methods of reheating are: (1) the air to be heated is passed through a cast-iron chamber or coil of pipe, exposed to a fire or current of hot gases or steam; (2) heat is added within the body of air itself, by the combustion of fuel, the injection of steam or hot water, or placing in the air pipe an electric-resistance coil. The first method is preferable from a mechanical standpoint and is the most efficient. The others may be employed where the ordinary burning of fuel is not admissible.

The following calculation * shows the theoretical results of reheating:

Weight of 1 cu.ft. of steam, at 75 lbs. gage = 0.2089 lb.

Total heat units in 1 lb. of steam, at 75 lbs., produced from water at 60° F. = 1151.

Total heat units in 1 cu.ft. of steam at 75 lbs. = 1151 × 0.2089 = 240.44.

* Frank Richards, "Compressed Air," p. 158.

To produce by compression in a steam-actuated air compressor 1 cu.ft. of compressed air at 75 lbs. gage and 60° F., about 2 cu.ft. of steam at the same pressure are required,* making the thermal cost of 1 cu.ft. of compressed air, at the above temperature and pressure, $240.44 \times 2 = 480.88$ heat units. The air is here supposed to have lost its heat of compression by being stored or transmitted to a distance, so that the 480.88 heat units represent its cost at the motor where it is used.

Result of reheating:

Weight of 1 cu.ft. of air at 75 lbs. and 60° F. = 0.456 lb.

Units of heat required to double the volume of 1 lb. of free air at 60° F. = 123.84.

Units of heat required to double the volume of 1 cu.ft. of compressed air at the above temperature and pressure = $123.84 \times 0.456 = 56.47$.

Comparing the thermal cost of 1 cu.ft. of air compressed in a cylinder with that of 1 cu.ft. obtained by reheating:

$$480.88 : 56.47 :: 1 : 0.1174$$

that is, the cost in heat units of a volume of air produced by reheating is less than $\frac{1}{8}$ of that required to produce the same volume by compression.

This theoretical result cannot be attained in practice. To effect such a saving a perfect reheater would be necessary, and the air after reheating would have to pass directly into the motor cylinder. A farther conveyance of the air in pipes for even a short distance rapidly lowers its temperature and pressure.

Reheating Temperatures. At constant pressure the volume of air is proportional to its absolute temperature, or 459° F. plus the sensible temperature above zero. The absolute temperature of air at 70° F. is $459 + 70 = 529°$. In doubling the volume by the application of heat the absolute temperature becomes 1058°, and $1058 - 459 = 599°$, which is the corresponding thermometric temperature. But this temperature is greatly reduced by the time the air reaches the motor cylinder, and still more heat is lost in the cylinder before its work is done. To reheat

* That is, the efficiency of the compressor is assumed to be 50%.

the air to a temperature which would double its volume in the motor cylinder would require reheating to a temperature much higher than 599°. If the temperature be raised by the reheater to 400° F. a loss of, say, 100° should be allowed for cooling before the air is actually used. The absolute cylinder temperature is then $300+459=759°$, and the added volume of compressed air practically available is:

$$529 : 759 :: 1 : 1.43+.$$

That is, by reheating to 400° F., there has been an effective increase of about 43% in the volume of compressed air. For proper lubrication, a higher temperature would be undesirable in the motor cylinder, and no material increase in economy could be realized in the operation of the motor. In practice the gain from reheating is considerably less than is here shown. For machine-drills and small, single-cylinder pumps, taking air at nearly full stroke, the increase of work ranges from, say, 30–35%, without deducting the cost of the fuel used in the reheater. Expansive-working engines show a higher efficiency.

For some purposes the determination of the quantity of heat to be imparted in reheating is based on the temperature at which the air leaves the compressor cylinder, the idea being to recover the heat subsequently lost in cooling. Suppose, for example, that the compression is practically adiabatic, as is usual in single-stage compressors. Taking as the unit 1 lb. of air, or 13.2 cu.ft., at 65° F., and compressing to 70 lbs. gage, the heat of compression is:

$$T' = T\left(\frac{P'}{P}\right)^{0.29} = 65° + 459°\left(\frac{70+14.7}{14.7}\right)^{0.29} = 869° \text{ absolute,}$$

and the final thermometric temperature is, $869° - 459° = 410°$ F. The rise in temperature due to compression is therefore:

$$410° - 65° = 345° \text{ F.}$$

If the compressed air be subsequently cooled to 65°, its volume becomes: $\frac{14.7 \times 13.2}{84.7} = 2.29$ cu.ft. In using this air

without reheating and non-expansively, in a rock-drill having 10% clearance, the work done is:

$$W = (2.29 \times 144 \times 84.7 \times 0.9) - (2.29 \times 144 \times 14.7) = 20,290 \text{ ft. lbs.}$$

But if the air be reheated to the final temperature of compression (345° F.), the work is:

$W' = \frac{869^\circ}{524^\circ} \times 20,290 = 33,478$ ft. lbs., and the gain by reheating is therefore: $33,478 - 20,290 = 13,188$ ft. lbs., or 39%.

The thermal cost of reheating this air will be: $345^\circ \times 0.2375$ (specific heat of air at constant pressure) $= 81.9$ thermal units (B. T. U.), equivalent to $81.9 \times 778 = 63,718$ ft. lbs. of work. Hence the efficiency of reheating in this case is:

$$13,188 \div 63,718 = 20.7\%.$$

A working test, by Prof. Alex. B. W. Kennedy, on a reheater supplying air for a 10-H.P. motor, gave the following results: The air was reheated to 315° F., by burning about 0.39 lb. coke per indicated H.P.-hour, producing an increase of about 42% in the volume, and, if the efficiency had remained the same as during the trials with cold air, there should have been a decrease of air consumption in the ratio $1 \div 1.42 = 0.7$. The volume of cold air used (admission temperature, 83° F.) was 890 cu.ft. per H.P.-hour; the volume when reheated was 665 cu.ft., or 75%; so that the full economy from reheating was nearly realized.

A summary of the results from two experiments on the above plant with cold, and two with reheated, air show:*

1. With cold air. Incomplete expansion, wire-drawing, and other such causes, reduced the actual horse-power of the motor from 0.50 to 0.39.

2. By heating the air to about 320° F. the horse-power at the motor was increased to 0.54. The ratio of gain due to reheating was therefore $0.54 \div 0.39 = 1.38$.

3. Deducting the value of 0.39 lb. coke per H.P.-hour, used

* "Experiments upon the Transmission of Power by Compressed Air in Paris." Van Nostrand's Science Series, No. 106, p. 35.

in heating the air, the net efficiency becomes 0.47, instead of 0.54, and the ratio of gain is reduced to:

$$0.47 \div 0.39 = 1.205.$$

These experiments, though not made with a well-designed reheater, show a substantial net gain from reheating. Where reheating is employed in mines, however, the quantity of heat imparted to the air is usually much less than that indicated above. Good results may be obtained by the application of even less than 100° F.

The results of experiments by Riedler and Guttermuth, on the consumption of reheated air by an ordinary single-cylinder 80 H.P. engine, are given in Table XXVIII.* This engine,

TABLE XXVIII

Test	Temperature of Air		Consumption Free Air per H P.-Hr. in Cu ft	Indicated Horse-Power.	Efficiency
	Admission, Deg F	Discharge, Deg. F			
1	264 2	69 8	462 77	72 3	0 89
2	305 6	84 4	431 09	72 3	0 90
3	320 0	95 0	418 55	72 3	0 91
4	338 0	120 2	432 12	65 0	0 87

with Corliss valve gear, was originally designed and used as a steam engine, and no changes were made for adapting it to work with compressed air, except that the cylinder was jacketed by the hot air on its way to the valve chest. The initial pressure was 95.5 lbs. absolute and the temperature of the air in the reheater did not exceed 338° F., at a coke consumption of 0.176 lb. per H. P.-hour.

Construction and Operation of Reheaters. The reheater employed in the experiments referred to in the preceding paragraph consisted of a double cylindrical box of cast iron, 21 ins. diameter by 53 ins. high, enclosed in a sheet-iron casing. The air traversed the annular space between the cylinders, being compelled by baffle-plates to circulate so as to come into con-

* Wm. Cawthorne Unwin, *ibid.*, p. 104.

tact with the whole heating surface. The products of combustion from a coke fire in the inner cylinder passed downward over the exterior surface of the annular air chamber on their way to the chimney. This served for a 10–12 H.P. motor. In another form of reheater the air passes through a coil of wrought-iron pipe, enclosed in a cylindrical casing. This gives a large heating surface, but wrought-iron pipe burns out rapidly unless the fire is kept moderate. Cast iron is preferable.

FIG. 126.—Ingersoll-Rand Reheater.

The Ingersoll-Rand reheater (Fig. 126) consists of two concentric cast-iron shells, one within the other, the joints being packed with asbestos gaskets. The inner chamber forms the top of the fire-box. In shape this reheater is a truncated cone, set on a cylindrical fire-box, the cold-air main being connected by a flange coupling at the top and the hot air discharged near the base. Dimensions: 42 ins. outside diameter at the base by

54 ins. high, with a grate 19 ins. diameter. It is stated that 340 cu.ft. of free air per min., at 40 lbs. pressure, can be heated to 360° F., with a gain of 30–35% in the energy developed. To reheat more than 400 cu.ft. of free air per min., 2 or more heaters are set in series, the air passing from one to another, allowing a maximum of 400 cu.ft. for each.

The inner and outer shells of cast-iron reheaters are subjected to considerable differences of temperature, and the upper and lower ring joints between the shells are often difficult to keep tight. In the old Rand reheater (Fig. 127), the castings are more complicated in shape, the air passing between them in a thin sheet, from the inlet on the side to the discharge at the top of the dome. To provide for expansion and contraction, the joint above the fire-box is made by a stuffing ring and packing.

The Sullivan reheater (Fig. 128) consists of a vertical coil of cast-iron piping, or hollow rings, encased in double sheet-steel shells, the space between the latter being filled with asbestos. Below is the grate and combustion chamber, the gases from which pass through the spaces between the air rings. To minimize leakage, the centers of the rings are joined by malleable-iron nipples, so that all expand and contract together. These heaters, usually designed for burning coal, coke, or wood, are made in 3 sizes, for 345–800 cu.ft. of free air per min., having from 3–7 rings, and measuring from 5 ft. 8 ins. to 7 ft. 6 ins. in height, by 33 ins. outside diameter.

Internally fired reheaters, in which the air is heated by direct contact with the fire, are unsuccessful, because dust and injurious products of combustion are carried into the cylinder of the air motor. This trouble does not exist to the same extent when gasolene or other oils are used, instead of solid fuels, and not at all in the electric reheater, which, however, has rarely been used. Most reheaters have no provision for regulating the heat according to the variation in consumption of air, as in running machine drills, channellers, hoists, and other intermittently operating engines. This want of regulation is less important for constant running engines, like pumps.

As the air chamber, in all externally heated or "dry"

reheaters, forms in reality a part of the air main, reheating increases the *pressure* only in a small degree. Its real effect is to increase the *volume* of air, which backs up in the main, reducing incidentally the velocity of flow and therefore the loss of pressure due to friction. The reheater should be placed as close as possible to the machine using the air. This is readily

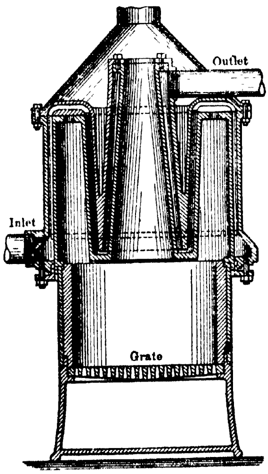


FIG. 127.—Old Rand Reheater.

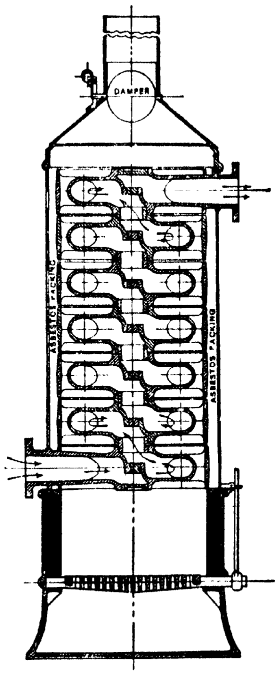


FIG. 128.—Sullivan Reheater.

done with stationary engines, and even in the case of movable machines, like quarry channellers, the reheater may be set on the same carriage.

With compound air-engines the exhaust from the high-pressure cylinder is sometimes passed through a reheater before going to the low-pressure cylinder. A further benefit may be

derived by injecting into the reheater a very small quantity of water. The specific heat of water is higher than the specific heat of air; also such part as is converted into steam gives up its latent heat in the air-engine cylinder and prevents trouble from freezing, even with a high rate of expansion. Similarly, it is beneficial to inject a little warm, or even cold, water into the feed-pipe of an air motor. Water thus used acts incidentally as a mechanical scourer, in removing accumulations of ice from the ports.

It is evident from the construction of reheaters that the calorific power of the fuel burned in them is not economically utilized. The flue loss is large for the same reasons that apply to the work of ordinary shell or tubular boilers. But the thermodynamic advantage gained is so marked that the low efficiency of the reheater itself, in burning the small quantity of fuel required is of secondary importance.

Reheaters for Underground Work. In the ordinary operations of mining the reheating of compressed air has a limited application. For portable machines like rock-drills, continually being shifted from place to place, the use of reheaters is economically out of the question, because they would have to be moved about with the drills, to prevent the reheated air from losing its heat and temporary increase of volume.

For stationary engines, however, as underground pumps, hoists and rope-haulage engines, where the reheater can be placed permanently close to the engine, reheating may be applied with a decided gain in efficiency. Incidentally it prevents the accumulation of ice in the exhaust ports. It may be difficult underground to arrange for burning fuel under a reheater, notwithstanding the small quantity required, because of the vitiation of the mine atmosphere. Where the conditions are such that strong combustion is not allowable, it will still be found that some advantage is obtainable from air engines by a very slight added temperature—say, only 25°–50° F. The use of the internal electric reheater, already referred to, in which a resistance coil is placed in an enlarged section of the air main, avoids the difficulty of disposing of the products of combustion

of fuel and would be especially useful in gassy collieries. Another mode of applying electric reheating is to wrap the resistance coils around a short length of the air pipe.

At the North Star Mine, Grass Valley, Cal., a reheater has been placed on the surface near the shaft mouth and the compressed air carried underground by a pipe covered with non-conducting material. While some saving can thus be realized for moderate distances, it is not practicable for long pipe lines. In any case, non-conducting covering should be used for the pipe from reheater to air engine, however short the distance. In a case on record,* where this distance was only 20 ft., without pipe covering, the gain in power was only 12%, though the absolute temperature of the air at the reheater was increased 38%, with corresponding theoretical increase of volume. For operating an underground pump in another California mine, the air is reheated by steam conveyed from the surface. Steam may thus be used to greater advantage than if employed directly in the cylinder of a pump; for, in condensing, the latent heat raises the temperature of the air and is so converted into work.

Reheating is important in connection with the operation of surface or underground hoisting engines by compressed air (see latter part of Chap. XVII).

Following are some results of experiments by Prof. J. T. Nicholson, in reheating air from the Taylor Hydraulic Air Compressor, at Magog (Chap. XV). The air was used in a 27-H.P. Corliss engine, at a pressure of 53 lbs. There were 5 tests: (1) with cold air; (2) reheating by steam injected into the air main near the engine; (3) the compressed air was passed into a steam boiler, and heated by mixing with the water and steam; (4) the compressed air was blown upon the surface of the water in the boiler, and heated by mixing with the steam; (5) the air was passed through a tubular reheater, fired by coke.

Without reheating, 850 cu.ft. of free air were used per I.H.P.-hour. By reheating in the boiler, a mixture of 10–15 lbs. of steam with the air reduced the consumption of air from 850 cu.ft. to from 300–500 cu.ft. per I.H.P.-hour. Thus, 1 added

* Richards, *American Machinist*, Feb. 28, 1895.

H.P. was obtained by *wet heating*, at an expenditure of 1–1.3 lbs. of coal per H.P.-hour. By *dry heating* in the coke-fired reheater, the air was raised to 287° F. At this temperature, 640 cu.ft. of free air were required per H.P.-hour, or 210 cu.ft. less than with cold air, the saving in quantity of air being about 25%. By burning in the reheater 47 lbs. coke per hour, 100 H.P. in cold compressed air was raised to 133 H.P., making an expenditure of 1.42 lbs. coke for each added H.P.-hour. Though these results indicate that the reheater used was not very efficient, the fuel consumption is far lower than is attainable in the best boiler and engine practice.

In a paper by Clarence R. Weymouth, on "Reheating Compressed Air with Steam," a detailed discussion is given of the thermodynamics of this mode of procedure, with deductions as to its efficiency. The author gives the results of injecting steam into the air main, and of passing the compressed air into a boiler.

CHAPTER XX

COMPRESSED AIR ROCK-DRILLS *

THIS chapter will be devoted to the description of a number of representative drills, together with notes on the performance, air consumption and operation of machine drills. For any rock harder than soft shales, coal, and other similar material, the percussion drill only is of practical use. All attempts to construct a rotary pneumatic rock-drill have thus far failed. The diamond, and other core, drills for deep boring, and the Brandt rotary drill, operated by hydraulic power, have no place in the present discussion.

General Description. The reciprocating or percussion rock-drill, except those machines that operate on the hammer principle (Chap. XXI), consist of a cylinder, in which compressed air or steam is used, the drill bit being clamped to a chuck on the end of the piston rod (Fig. 129). Many different valve motions are in use. Some resemble the valve motions of certain single-cylinder pumps; in others the valve is positively moved by a tappet, actuated by the strokes of the piston. The necessary rotation of the drill bit on its axis, for keeping the hole round and preventing the bit from sticking (" fitchering "), is produced automatically by a rifle-bar, ratchet and pawls. Working speeds are from 300–400 strokes per min. for the larger sizes of drills, to 500 strokes for the smaller; normal length of stroke, in drills of average size, 4½–6 ins. The admission of air, and therefore the speed and force of the blow, are controlled by a hand valve in the air pipe close to its connection with the valve chest.

A feed screw, with crank and handle, is carried in a bearing

* Except as regards drill mountings, this Chapter deals with Reciprocating Drills (see General Classification, p. 273). For Hammer Drills, see Chap. XXI.

at the rear end of the shell supporting the cylinder, and engages with a nut on the under side of the cylinder casting. The entire drill head is thus fed forward by hand as the hole is deepened. (An automatic feed, used to some extent for surface work, is neither necessary nor satisfactory for underground service.) When the cylinder has been fed forward as far as the length of the screw and of the supporting shell will permit, the drill is stopped. By reversing the feed the cylinder is moved back on the shell, the bit is removed and a longer one clamped in the chuck. The cylinder is then fed forward until the new bit touches the bottom of the hole, with the piston nearly at mid-stroke; the air is turned on slowly and the work proceeds. The length of stroke, and therefore the force of the blow, are under the control of the drill-runner. If the drill is fed forward faster than the hole is being deepened the stroke necessarily shortens, because the bit strikes the bottom of the hole before the full length of stroke is reached; conversely, with too slow a feed, the piston will strike the front cylinder head. Thus, the force of the blow may be regulated to suit the conditions. When starting a hole, the stroke should be short until the bit has adjusted itself to the shape of the bottom of the hole. For hard rock, a short, rapid stroke is best; a longer stroke for softer or tough rock.

Drill Mountings. The drill head, comprising the cylinder and its appurtenances and the supporting shell, is mounted on a tripod or column. For surface work the tripod only can be used; underground, either the tripod or column, depending on the size and shape of the working in which the drilling is to be done.

1. *Tripod.* (Fig. 129). The legs, which are telescopic, are hinged by a heavy bolt to the tripod head, and can thus be set as necessary for adjusting the position of the axis of the cylinder and bit, for the hole to be drilled. To the tripod head is bolted the "shell," which has guides for supporting the cylinder, as it is fed forward. After the machine is in position for drilling all bolts are tightened. Weights are usually slipped on the tripod legs, to prevent the drill from shifting while in operation.

Tripod mountings are required for underground work, when the distance between roof and floor, or between the walls of the

workings, is too great to permit the use of columns. Where a choice exists, the tripod is sometimes preferred to the column,

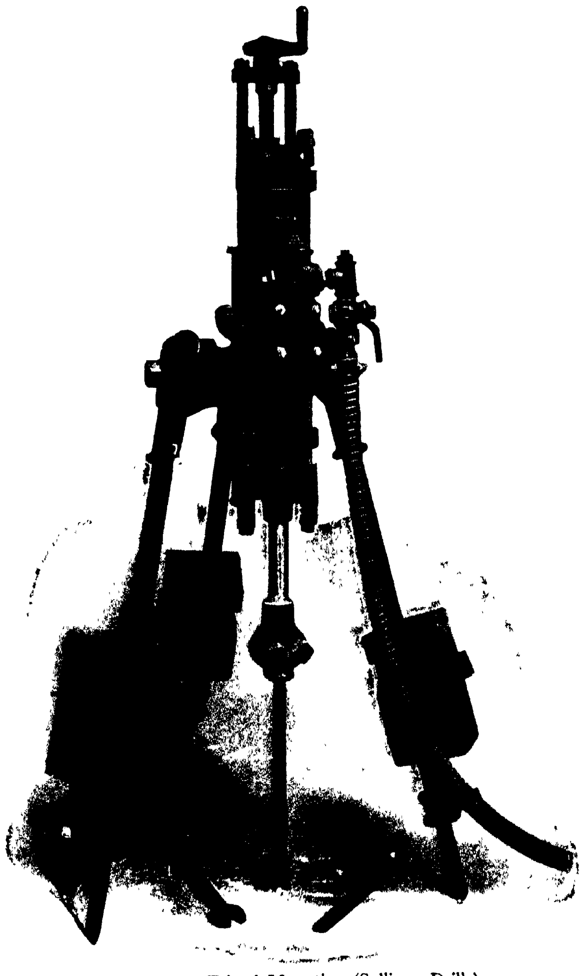


FIG. 129.—Tripod Mounting (Sullivan Drills).

because it can usually be set up in less time and allows greater freedom in locating the holes, so as to produce the best results

under existing conditions as to character of rock and shape of the face. This may be true, also, in sinking shafts of large cross-sectional area, or when the rock is so irregularly fissured that a symmetrical arrangement of the round of holes is undesirable.

2. *Column or Bar* (Fig. 130) is a heavy steel tube, 3–5½ ins.

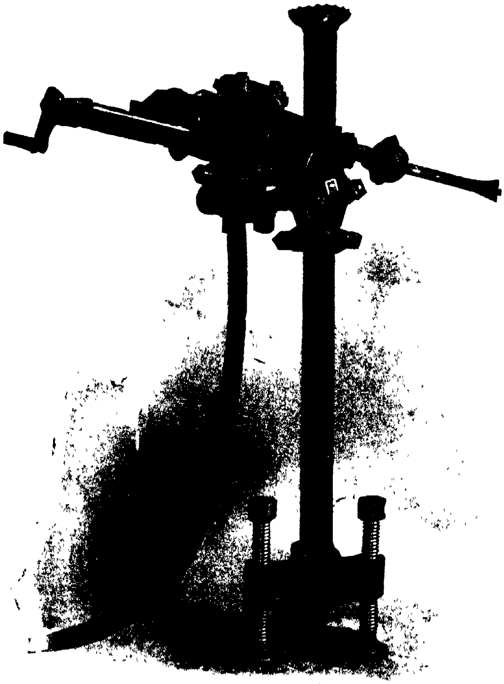


FIG. 130.—Double-Screw Column Mounting.

diameter, according to the size of drill. It is set between the roof and floor, or the walls, of the working. The lower end of the column shown has a cross-piece or base, through which pass a pair of jack-screws. The upper end has a serrated head, which, by tightening up the screws, takes a firm hold upon the surface

against which it is pressed; blocking and wedging are generally required. Another form of column, with a single, telescopic jack-screw, is useful in small tunnels or mine workings, and for shaft-sinking. In large workings the double-screw column is preferable. The drill is carried on an arm, attached to a collar which is clamped fast in any desired position, for adjusting the height and angular position of the drill. With the single-screw column the drill is attached directly to the collar, the arm being omitted. Sometimes two drills are mounted on the same column.

The column mounting is best for driving mine tunnels, cross-cuts, and drifts, and for advance headings of railroad tunnels. It is often employed, also, for stoping in veins less than about 7 ft. thick, when the dip of the vein and the method of mining make it difficult to use tripods. When set horizontally, as in shaft-sinking, the column is called a "bar." Shaft-bars of extra length, for wide shafts, have a pair of adjustable legs hinged at the middle point to carry the weight of the drill or drills without making the bar inconveniently heavy.

Formerly, for driving railroad tunnel headings, 4, 6 or more, drills were mounted on a carriage on a track, but they are now rarely used.

General Classification of Rock-Drills. (*A*) Reciprocating drills, comprising those in which the drill bit is firmly attached to the piston rod; (*B*) Hammer drills, in which the bit does not reciprocate, but is held in a socket in the forward end of the cylinder, and is struck by the piston as by a hammer.*

RECIPROCATING DRILLS

Classification based on the design of the valve-motion:†

A. Air-thrown valve machines (nearly all are of the spool-valve type).

B. Tappet-valve machines.

C. Miscellaneous types.

Examples of each form are given below.

* Air Hammer drills are discussed in Chap. XXI.

† For another classification, see *Eng. & Min. Jour.*, Feb. 22, 1913, p. 421.

AIR-THROWN VALVE MACHINES

"Sergeant" Drill. Fig. 131 is a longitudinal section. The spool-valve and main air and exhaust ports are somewhat similar to those of a single-cylinder pump. Air is admitted on one side of the valve chest, the exhaust opening being on the other side.

The valve-motion consists of two parts: a spool-valve, which controls the main cylinder ports, and an arc-shaped tappet, set in a curved slot or seat, between the cylinder and valve chest. The ends of this tappet, projecting slightly into the main cylinder, are struck alternately by the front and back shoulders of the large annular recess in the middle of the piston, thus causing the tappet to oscillate at each stroke of the piston. Alongside of the tappet, and in its seat, are three auxiliary ports, one in the middle and one near each end of the seat. These ports connect with the spool-valve chest above; the middle port with the middle of the chest, the rear port (*i.e.*, nearest the back end of the cylinder) with the *forward end* of the chest, and the forward port with the *rear end* of the chest. In the face of the tappet is a curved slot, just long enough, when in the extreme positions of its throw, to form a communication between the middle auxiliary port and one of the end auxiliary ports. Therefore, at each stroke of the piston, the tappet places the *opposite end* of the valve chest in communication with the exhaust, thus causing the throw of the spool-valve. In the peripheral surface of the spool-valve is a very fine longitudinal slot, which constantly admits a small quantity of live air to both ends of the chest. Hence, when either end of the chest is connected with the exhaust, the valve is thrown towards that end by the air pressure in the other end of the chest.

In Fig. 131 the piston is beginning its forward stroke. The spool-valve, in its rear position, is admitting air to the back end of the cylinder, while the forward end is connected with the exhaust. As the piston advances, the rear shoulder of the annular recess in the piston strikes the projecting end of the tappet and throws it over. This changes the relation between

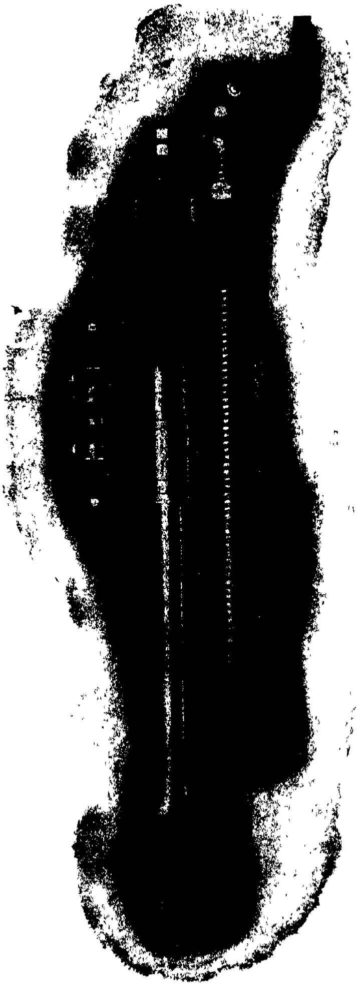


FIG. 131.—"Sergeant" Rock Drill Ingersoll-Rand Co.

the auxiliary ports, described above, exhausts the air from the front end of the chest and throws the spool-valve forward, thus preparing for the back stroke of the piston.

The use of the arc-tappet avoids in large part one of the defects of the ordinary spool-valve drills, *viz.*, irregularities in the operation of the drill, caused by wear of the valve and seat, which permits leakage of air past the valve. Adjustments for irregularities of stroke produced by wear are made by the compensating device shown in Fig. 132, which is an enlarged section of the spool-valve and chest. A hollow brass plug *P*, having a very small hole *H*, permits the passage of a little live air to the back end of the chest. Should the piston strike the back cylinder

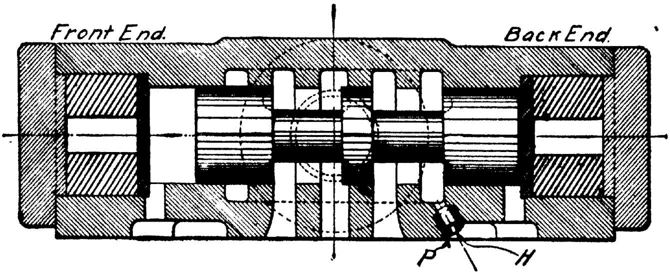


FIG. 132.—Spool-Valve and Chest. "Sergeant" Rock Drill.

head, the area of *H* is reduced slightly by peening (hammering) up the edge of the hole. This decreases the quantity of air passing to the end of the chest, throws the valve a little later, and so increases the cushioning in the rear end of the cylinder. If the stroke be too short, *H* may be found partly obstructed and should be cleaned, to admit more air to the end of the chest; if the stroke be still too short, the area of *H* is slightly increased by reaming out with the point of a knife blade.

Rotation of piston and bit: A rifled bar, with a ratchet head and pawls set in the rear cylinder head, engages with a correspondingly rifled nut screwed into the end of the hollow piston. The ratchet teeth and pawls (Fig. 133) are so shaped that, on the forward stroke, the piston moves without rotation, the

rifle-bar turning the ratchet in its seat. On the back stroke the pawls prevent the ratchet from turning, so that the piston is compelled by the rifle-bar and nut to rotate through a part of a revolution. The ratchet ring, with internal teeth, with which the pawls engage, is not fastened rigidly in the back cylinder head, but is held by

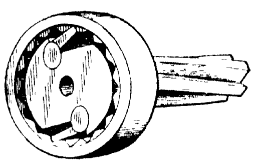


FIG. 133.—"Sergeant" Ratchet and Rifle-Bar

friction only, under pressure of a cushion spring in the back head of the cylinder. Hence, when the drill bit sticks in the hole, or for any reason cannot rotate freely on the back stroke, the ratchet itself turns, thus preventing injury. The drill is fed forward on its supporting shell by a long feed screw, engaging with a feed nut in a lug on the under side of the cylinder.

The Sergeant drill is built in seven sizes, the cylinders measuring: 2, $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, 3, $3\frac{1}{4}$

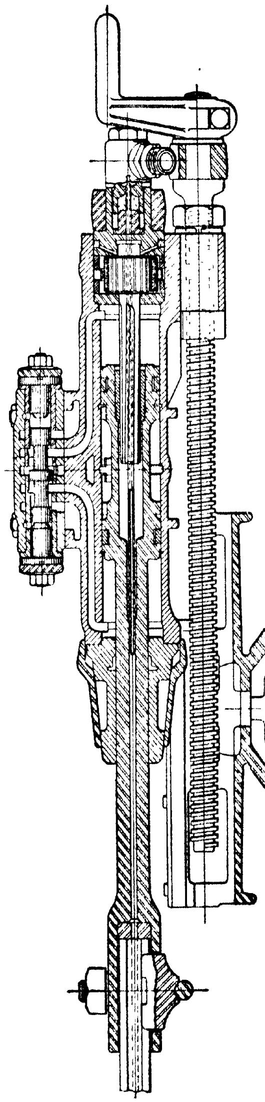


FIG. 134.—"Liteweight" Drill, with Water Attachment. Sullivan Machinery Co.

and $3\frac{1}{2}$ ins. diameter; weights of drillhead, unmounted, 110–405 lbs.

Sullivan "Liteweight" Drill (Fig. 134) has a three-ringed spool valve, thrown by air admitted from and exhausted directly into the cylinder, through auxiliary or reverse ports. These ports (not fully shown in the cut) are grooved in the valve-chest, instead of being cored in the cylinder casting, as in the older pattern of the drill. The middle ring of the valve is of larger diameter than the others. As the piston makes its stroke, the reverse ports put the ends of the chest into communication with *opposite* ends of the cylinder, thus throwing the valve and preparing for the next stroke of the piston. The valve chest is bushed with steel. The valve will still reverse even when the

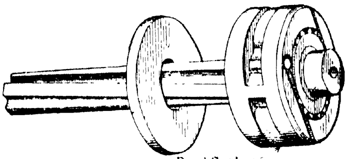


FIG. 135.—Rotation Device, Sullivan "Liteweight" Drill.

drill head is fed down so that the piston stroke is shortened to only about $\frac{1}{2}$ in. A steel liner, $\frac{3}{32}$ in. thick, is pressed into the cylinder and dowelled to prevent movement. It gives a hard, smooth surface to take the piston wear, and enables a lighter cylinder casting to be used. An automatic lubricator is provided at the rear end of the cylinder.

The ratchet of the rotation device (Fig. 135) has rounded toothed surfaces, engaging with two small steel rollers, instead of angular pawls. These rollers are held in place by peripheral springs, as shown. The ratchet ring and collar are held against the rear cylinder head by friction only, to permit slippage in case the bit wedges in the drill hole.

This drill is made both with and without a water injection attachment, for keeping the bottom of the hole clean by driving out the sludge and thus increasing the drilling effect. Fig. 136

shows the water type of drill.* The piston, piston rod and rifle-bar are bored out to permit insertion of a small tube, through which water and compressed air pass under pressure into the hollow bit to the bottom of the hole. The water supply is from an 18-gal. tank, connected by hose to the compressed-air pipe and to the drill. A gravity supply of water may replace the tank, provided the water pressure does not exceed the air pressure. Incidentally, the use of the water attachment saves the time ordinarily required to clean out the drill hole at intervals.

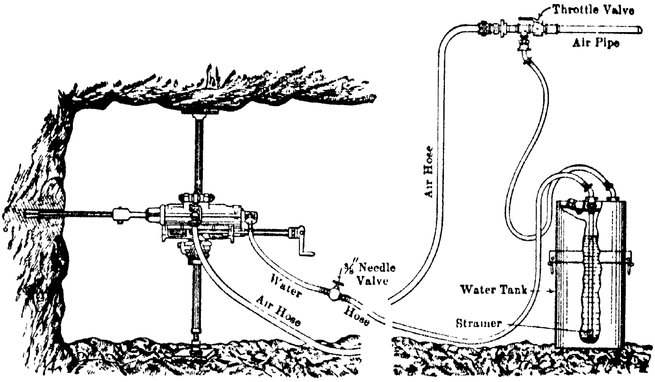


FIG. 136.—Sullivan Drill Mounted, with Water Tank and Connections.

The "Liteweight" drill is made in 3 sizes: $2\frac{5}{8}$, $3\frac{1}{4}$ and $3\frac{5}{8}$-in. diameter of cylinder. The $2\frac{5}{8}$-in. size is a one-man machine, weighing unmounted, 162 lbs.

Wood Drill (Fig. 137) is another example of spool-valve machine. With a few exceptions, its main features are similar to those of the Sullivan drill, already described. The front and back vertical reverse ports for the valve are bushed, as shown, and, by means of horizontal channels cut in the lower part of the valve-chest casting, communicate with opposite ends of the chest. By the reciprocations of the piston the ends of the

* It should be stated that this device has been used for some years in the "Water-Leyner," now, the "Leyner-Ingersoll," drill (Chap. XXI).

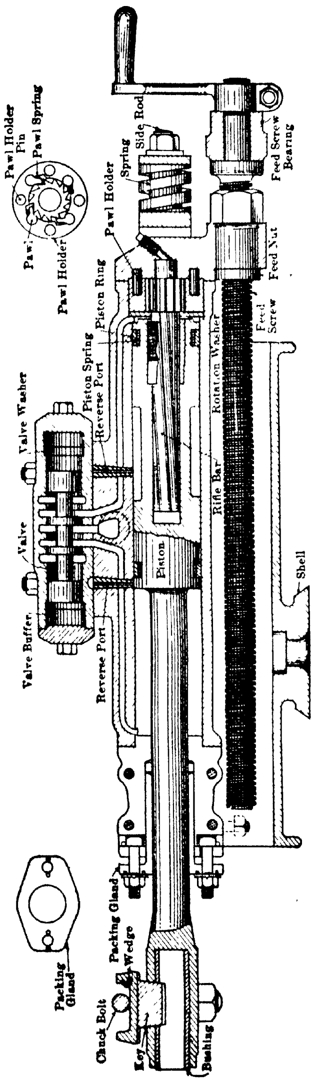


FIG. 137.—"Wood" Piston Drill Wood Drill Works.

chest are thus connected with the cylinder, and thence through the main ports back to the chest and finally to the exhaust opening of the machine. The rotation of the piston and bit are positive, the ratchet having no slip ring. The four pawls are set in a solid forging (pawl-holder), which is held rigidly by dowel pins, as shown, between the pawl-holder and rear cylinder head.

This drill is made in 6 sizes for underground service: 2, $2\frac{1}{4}$, $2\frac{3}{4}$, 3, $3\frac{1}{4}$ and $3\frac{5}{8}$-in. diameter cylinder; weights, unmounted, 85–390 lbs. The two smallest sizes are one-man machines.

Climax Imperial Drill (Fig. 138). This is a well-known English machine. Air enters the valve-chest by the air tap, and thence passes into the annular recess *b* of the valve, which, by its reciprocations, opens communication through the valve-seat ports *c*, alternately with the main cylinder ports *a,a*. In

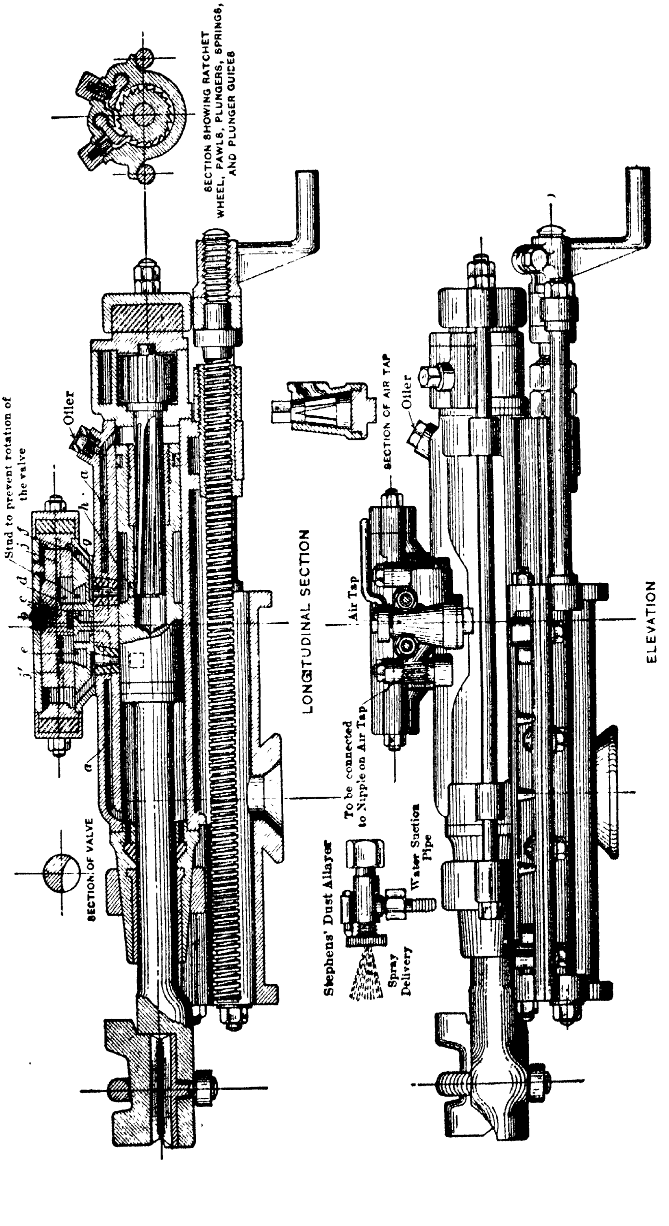


FIG. 138.—Stephens' Climax Imperial Drill, 3¼-in. R. Stephens & Son, Carn Brea, Cornwall, England.

the valve are the recesses *d*, shown also in section to the left. These, by the movements of the valve, control the exhaust ports *e*, which connect with the main exhaust on the side of the chest. The valve is thrown by admitting a little live air through the small grooves *j,j'*, to the ends of the chest, this air being alternately discharged by the larger auxiliary ports *f,f'*. These open into the cylinder through the auxiliary ports *g*, exhausting at each stroke into the annular recess of the piston and thence into one of the square ports *h*, leading to the main exhaust. Ports *g* are bushed with composition metal rings, shaped at the lower end to fit closely upon the piston. As shown in the cut, the piston is in position to begin its forward stroke; the valve has been thrown to the right and is admitting air to the rear end of the cylinder. The drills are designed to run at the high speed of from 450–500 strokes per min.

A specialty of this drill is the "dust allayer," which is attached to the air tap by a nipple and cup, forming a ball-and-socket joint. It is, in effect, an ejector, drawing water from any convenient source, by means of a small quantity of compressed air led from the throttle. By the same air the water is sprayed forward into the mouth of the drill hole.

The "Climax" drill is made in 7 sizes: $1\frac{3}{4}$, 2, $2\frac{1}{4}$, $2\frac{1}{2}$, 3, $3\frac{1}{4}$ and $3\frac{1}{2}$-in. diameter of cylinder.

Holman Drill, an English machine, is made in $2\frac{1}{8}$, $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, $3\frac{1}{4}$, $3\frac{1}{2}$ and $3\frac{5}{8}$-in. sizes. In a few of their details the smaller sizes differ somewhat from the larger, though the general design is the same in all. Fig. 139 illustrates the sizes from $2\frac{1}{4}$ to $2\frac{3}{4}$-in. The movements of the spool-valve 1, which control main air and exhaust ports of the usual form, are caused as follows: Below each end of the valve-chest, and communicating from chest to cylinder, is a short vertical port, with a coned or taper seating. In each of these ports is a pair of steel balls 4, 5, the former of which controls the auxiliary port 7. Both balls are under the pressure of the spiral spring 8. The seat is so shaped that the lower and smaller ball 5 will project slightly into the cylinder, whenever permitted to do so by the position of the annular recess 6, around the middle of the piston. Hence,

by each piston stroke the lower ball 5 receives a slight upward blow from the inclined shoulder of the recess. This lifts the larger ball 4, thereby opening auxiliary port 7, and placing the corresponding end of the valve-chest in communication with the main exhaust 3. Owing to the pressure of the air occupying the opposite end of the chest, the spool-valve is then reversed, to prepare for the next stroke of the piston. Ball valves are not liable to breakage, and, as they receive a slight rotary motion from each blow of the piston, the wear tends to be equalized, thus keeping them round and preventing leakage between the balls and their seats.

Ingersoll-Rand "Butterfly" Drill (Figs. 140–143). The valve (Fig. 141) is "air-thrown," having no mechanical connection with the piston. It consists of two flat wings,

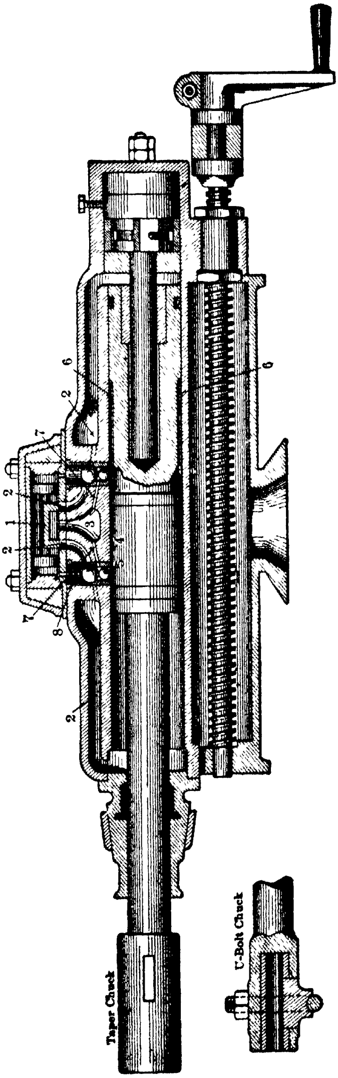


Fig. 139.—Holman Spool-Valve Drill (Camborne, Cornwall, England).

with ground seating surfaces, these wings forming one piece with a central trunnion. The valve oscillates slightly in a groove or slot in the chest, being actuated by the unbalancing of air pressure alternately on the wings. There are 4 ports, opening into the faces of the valve slot, a separate inlet and exhaust

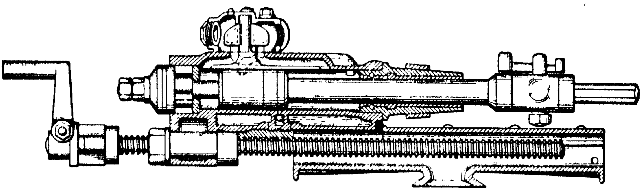


FIG. 140.—Ingersoll-Rand "Butterfly" Drill, for Tripod or Column Mounting.

for each end of the cylinder. The two inlet ports open opposite to each other at one end of the slot, the exhaust ports being similarly placed at the other end. When one of the valve wings closes the inlet to one end of the cylinder, the opposite face of the other wing closes the exhaust port from the other

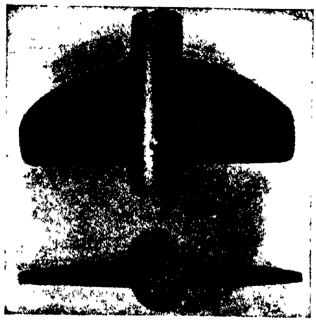


FIG. 141.—Valve of "Butterfly" Drill.

end of the cylinder. One inlet and one exhaust port are therefore always open.

Fig. 142 shows diagrammatically the operative conditions on beginning the forward stroke. The rear inlet port S_2 and the forward exhaust port E_1 are open, the others being closed.

Since the piston in this position covers the rear exhaust port E_2, the pressure of the inlet air holds the valve to its seat over port S_1. When the advancing piston uncovers E_2, live air passes through the cylinder to E_2, almost balancing the pressure on the two valve wings. But, the resultant of the forces acting on the valve, comprising the flow of the exhaust air in E_1, the impact of the inlet air on entering the chest, and the friction of the air in passing through S_2 and E_2, keeps the valve on its seat, in its first position, until the forward stroke is nearly completed. When the piston passes over and closes the forward exhaust port E_1, the cushion pressure produced in front of the piston acts through the port S_1, and throws the valve to the position shown by Fig. 143. The back stroke now begins, live air enter-

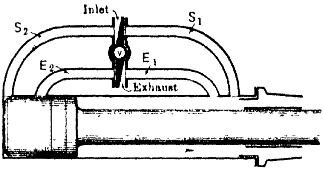


FIG. 142.

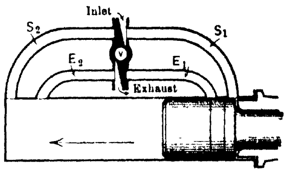


FIG. 143.

ing through S_1, while the exhaust escapes through E_2. The cycle of operation is then completed, as already described.

The "Butterfly" drill is a high-speed machine, making 500–600 strokes per min. The ports are large and with a small amount of movement the valve gives a large port area. Tests on the earlier drills of this type indicated that the air consumption is rather high. A new 2¾-in. drill showed a consumption of 95 cu.ft. free air per min. at 70 lbs. pressure and 125 cu.ft. at 85 lbs. This is due to the fact that, when the valve reverses, there is momentarily a direct communication from inlet to exhaust, as from S_2 to E_2. As the valve trunnions and faces wear, the seating may become imperfect, so that notwithstanding the quick action of the valve there is a loss of air. For example, a new 3-in. drill, tested at 70 lbs. pressure (with an Excelsior air meter), used 120 cu.ft. free air per min. The

same drill, with valves and valve-chests 3 months and 5 months old, consumed respectively 145 and 170 cu.ft. Later improvements have largely overcome these difficulties and lessened the air consumption.

This machine is a fast driller and marks a distinct advance in simplicity of construction; it "muds" well, and, being a one-man drill, the labor cost is low. The "Butterfly" valve is used also in the Leyner-Ingersoll and Ingersoll-Rand hammer drills (Chap. XXI).

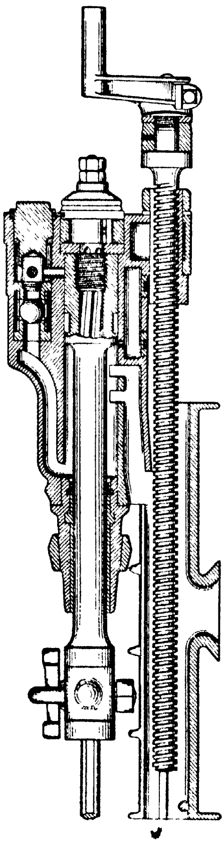


FIG. 144.—"Chicago Gatling" Drill. Chicago Pneumatic Tool Co.

"Chicago Gatling" Drill (Chicago Pneumatic Tool Co.) affords another example of an air-thrown valve, not of the spool type (Fig. 144). The valve is a 2-oz. hollow steel ball, held in a cylindrical pocket or cage, taking the place of the usual chest. Around the periphery of the cage are 12 small apertures for admitting air. The ball, with a $\frac{1}{8}$-in. throw, controls the main ports to the cylinder; seating surfaces are of steel. Live air pressure holds the ball on its seat until a drop in pressure on the other side of the ball is caused by the piston uncovering the corresponding exhaust port. That is, the valve movement is not dependent upon compression; the ball is thrown by the difference in total pressure on the opposite sides, the exhaust opening being larger than the apertures in the valve cage, through which live air enters. A similar valve motion is used in the Chicago "Hummer" drill (Chap. XXI).

Speed of stroke, 600–700 per min.; the stroke is uncushioned. Size of drill, 2¾-in.; weight, unmounted, 200 lbs.

Other Spool-Valve Drills are built by the McKiernan-Terry Drill Co., N. Y., the Chicago Pneumatic Tool Co., and the Cochise Machine Co., Los Angeles, Cal. The "Chicago Slogger" has an arc-shaped auxiliary tappet valve, resembling that of the "Sergeant" drill (Fig. 131).

The Siskol drill was one of the four successful competitors in the fourth (1909) Transvaal stope-drill contest, which was conducted underground, under regular working conditions, and lasted nearly a year. It is built in two sizes, 2$\frac{9}{16}$-in. and 2$\frac{15}{16}$-in. diameter of cylinder. Fig. 145 shows the one-man

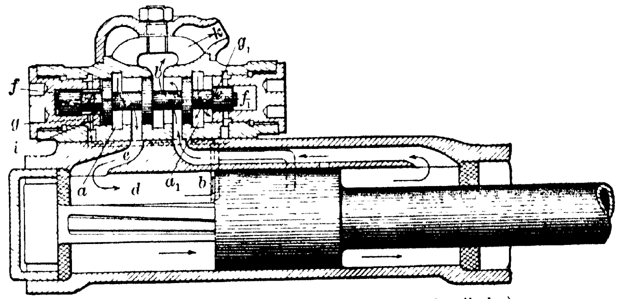


FIG. 145.—Siskol Drill (2$\frac{9}{16}$-in. diam. of cylinder).

2$\frac{9}{16}$-in. stoper, which weighs, unmounted, about 120 lbs. During the test run, with two of these machines, 14,083 linear ft. of hole were drilled in 215 8-hour shifts; average speed, 0.818 in. per min. per drill. The other successful competitors, Holman 2⅛-in. and 2¾-in. (Fig. 139), and Chersen (see below), also working in pairs, drilled respectively 12,779, 11,744 and 11,781 ft. of hole in the same time, average speeds, 0.742, 0.682 and 0.684 in. per min. per machine. In another test of the 2$\frac{9}{16}$-in. drills, in May, 1911, lasting 1½ hours, 14.06 cu.ft. of free air were used per linear in. of hole; air pressure, 75 lbs.

The Chersen drill is a 2¾-in. stoper, which also gave good results in several Transvaal contests. It is peculiar in having the spool valve chest placed transversely across the cylinder.

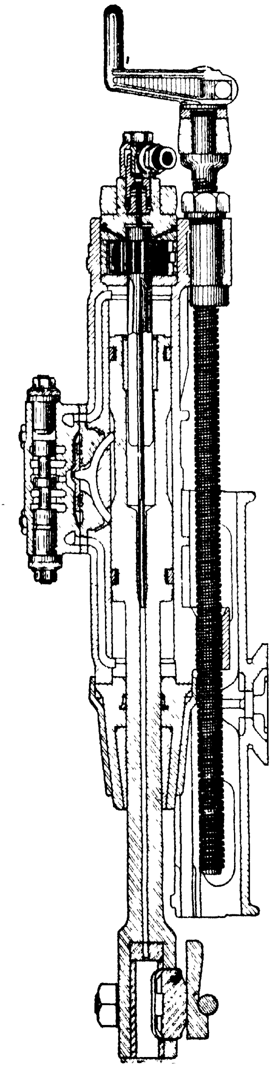


FIG. 146.—Sullivan "Hy-Speed" Drill with Water Attachment

The valve has a very short stroke—15 one-thousandths of an inch—and is actuated by a slight cushioning of the air in the cylinder, at each end of the stroke. Owing to its small travel, the action of the valve is apt to be interfered with by the entrance of grit into the chest. This difficulty was clearly brought out in the Transvaal tests.

TAPPET-VALVE MACHINES

Sullivan "Hy-Speed" Drill (Sullivan Machinery Co.) has a spool-valve which controls the main cylinder ports, and a small auxiliary slide valve, thrown by a 3-arm tappet or rocker (Fig. 146). The tappet oscillates in an arc-shaped slot, as its lower arms are struck alternately by the beveled shoulders of the piston. On the back of the tappet is an arm of standard rack-tooth form, which engages with a socket in the underside of the slide valve, thus throwing this valve positively. The slide valve controls small auxiliary ports, communicating with the spool-valve chest, thus causing the spool to reciprocate. An advantage of the flat valve is that it is held to its seat by air

pressure, so that wear does not cause leakage. The non-pivoted tappet here used is an improvement on the design used for many years for all tappet drills; breakage of the tappet is greatly reduced. Automatic lubrication is provided, as in the "Liteweight" drill.

The rotation, feed mechanism, and water attachment, are the same as in the Sullivan "Liteweight" drill (Fig. 135). Drill sizes, $2\frac{3}{4}$, 3, $3\frac{1}{4}$, $3\frac{5}{8}$ and $4\frac{1}{4}$-in. diameter of cylinder; weights, unmounted, 250–730 lbs.

The $3\frac{5}{8}$-in. size is made in two styles. The heavier (weighing 730 lbs.) is the "Engine-feed" drill, designed for drilling 3-in. holes to 20 ft. deep in quarrying and other open-cut work. Its valve-motion and rotation are as described above, but it is mounted on a special shell, to give a continuous feed of $4\frac{1}{2}$ ft. Due to its unusual weight, a small 2-cylinder hoisting engine is attached to the back of the shell for feeding and raising the drill. This engine is geared to the top of the feed screw, which also has a handle, for feeding by hand, if desired.

Murphy "Little Champion" Drill (Fig. 147). The rotation mechanism is similar to that of other drills already described.

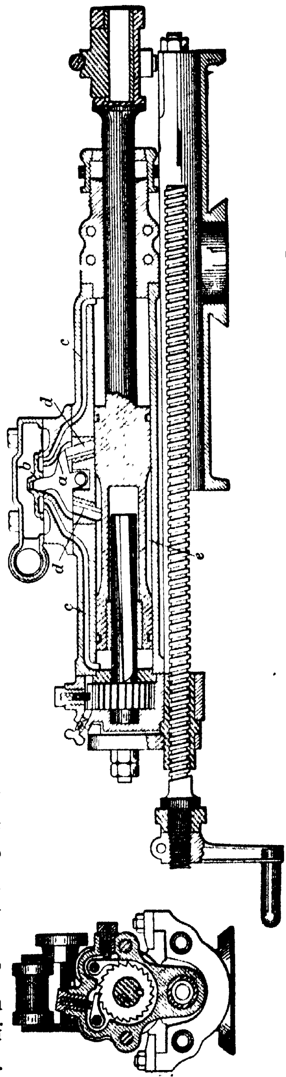


FIG. 147.—Murphy "Little Champion" Drill. Wickes Machinery Co.

The upper arm of the 3-arm tappet *a* engages with the flat slide valve *b*. This valve controls the main ports *c,c* and the exhaust port, which is opposite the middle of the tappet. A novel feature of the drill is the mode of producing the tappet's movements. Its two lower arms do not project into the top of the cylinder bore, to a contact with the piston, as is the case with most drills of this type. Instead, a steel pin *d*, sliding in a bushed opening in the cylinder casting, is set under each tappet arm. As the piston makes its strokes, the curved shoulders of the annular groove *e*, in the piston, alternately strike the rounded ends of the forward and rear tappet pins, *d,d*, thus pushing up the pins and causing the tappet to oscillate.

There are eight sizes: $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, 3, $3\frac{1}{8}$, $3\frac{1}{4}$, $3\frac{1}{2}$ and $3\frac{5}{8}$-in. diameter of cylinder, the drill head and shell weighing, unmounted, 125–395 lbs.

Holman Drill is an English tappet-valve machine, made in Camborne, Cornwall (see also Holman spool-valve drill). A 3-arm pivoted tappet, oscillated by shoulders on the piston, throws a *D* slide valve, which is held on its seat by a plate spring bearing against the valve-chest cover.

MISCELLANEOUS TYPES

Temple-Ingersoll "Electric-Air" Drill (Fig. 148) is unique in the mode of combining both systems of power transmission and in no way belongs to the class of electric-driven drills, which for years have been brought out from time to time, but which as yet have not given wholly satisfactory results.*

This machine comprises three parts: a drill, and an air pulsator driven by an electric motor. Pulsator and motor are mounted on a small truck, close to the drill and connected with it by two short lengths of hose. The drill differs materially from an ordinary rock-drill; it is valveless, and the cylinder is of larger diameter; the short piston, with packing rings, resembles the piston of a steam engine. The drill may be mounted on a column or tripod, and is provided with a feed screw and rotation

* Some resemblance to this machine is traceable in the design of the "Pneumelectric Coal Puncher" (Chap. XXII),

device. It has no buffers, springs or side rods. The pulsator is in effect a duplex, single-acting compressor, with cranks at

Fig. 148.—Temple-Ingersoll " Electric -Air " Drill.

180°, the crank-shaft being driven through single-reduction gearing from the motor armature. From the pulsator one hose

passes to the back end of the drill cylinder, the other to the forward end. These connections serve as ports for admission and return of the compressed air. There is no exhaust; the air circuit is closed, the same air being used over and over. Thus the speed of stroke depends on the motor speed, which is varied by a controller, operated through a cord by the drill runner. If a direct-current motor be used, it is designed for three speeds; or an alternating-current, single-speed motor may be employed if desired.

The pulsator runs at a low air pressure, only a small degree of compression being necessary for transmitting the power, the air acting as a spring between pulsator and drill. Incidentally, the air cushions the drill piston at the ends of the stroke. Leakage from joints and past the pulsator pistons is made up by a compensating valve (not shown in the cut), adjusted to maintain a practically constant pressure in the air circuit. When the pressure falls below the limit, the valve opens automatically and admits more air, which is compressed by the differential area between the two parts of the piston in the first cylinder, until normal pressure is restored. The pulsator cranks and pistons are lubricated by the "splash" method, the lower part of the crank-case being partly filled with oil. Some oil is atomized and carried with the air into the drill cylinder.

This drill is made in 4 sizes (Table XXIX), which, in working capacity, correspond approximately to the 2, $2\frac{3}{4}$, 3 and $3\frac{1}{4}$-in. sizes of the "Sergeant" drill (Fig. 131). The voltage recommended is 220. For alternating current the standard motors (which are stronger and simpler than those for direct current)

TABLE XXIX

No.	Diameter Cylinder, Ins.	Stroke, Ins.	Strokes per Min	Weight of Drill unmounted, Lbs.	Weight of Pulsator and Motor, Lbs.		Approx. H.P. at Pulsator for 1 Drill.
					D C.	A C.	
3-C	$3\frac{5}{8}$	$6\frac{1}{4}$	475	119	525	370	3
4-D	$4\frac{3}{4}$	7	415	223	645	545	4
4-E	$4\frac{3}{4}$	7	440	228	883	928	4
5-C	$5\frac{5}{8}$	8	400	299	1050	820	$5\frac{1}{2}$

are three-phase, 25, 30, 50, and 60 cycle. Direct-current motors, wound for 440 or 500 volts, may be used, but these voltages are unnecessary and somewhat dangerous for underground service.

"**Triumph**" **Drill** (Fig. 149) is valveless. Air is admitted by the two-way throttle *c*, on top of the cylinder, entering thence the annular port *d*. On beginning the stroke, as in the cut, *d* is in communication with an annular recess *e*, near the forward end of the piston; whence the compressed air passes through *f*, which is one of four longitudinal ports in the body of the piston, to the rear end of the cylinder. The forward stroke then takes place, the air in front of the piston being exhausted through ports *j*. As the piston advances, exhaust ports *j* are covered by the solid part of the piston, thus cushioning the end of the stroke. When the annular recess *e*, in the piston, comes opposite ports *j*, the air exhausts from the back end of the cylinder. At the same time, the annular recess *h*, near the rear end of the piston, comes into connection with the inlet *d*, thus admitting live air through the four longitudinal piston ports, one of which, *i*, is shown. These ports conduct the air to the forward end of the cylinder and the stroke is reversed. The parts are so proportioned that the air acts at full pressure throughout only about one-third of the forward stroke and then expands. Should the piston strike the cylinder head, the shock is absorbed by the spring *a*. In the front head is a stuffing box, the gland of which is held in place by the cap *b*.

Contrary to the usual design of rotation devices, the rifle-bar *k* is solidly screwed into the piston, engaging with the rifled nut *m*, in the ratchet *g*. The outer end of the rifle-bar extends into the closed tube *r*, which is connected with the cylinder by the small passages *x* and *y*. Hence, live air acts on the entire rear-end area of the piston, including the area of the rifle-bar. The tube *r*, and connecting passages, serve incidentally for the better distribution of oil on both sides of the ratchet and rifle-nut.

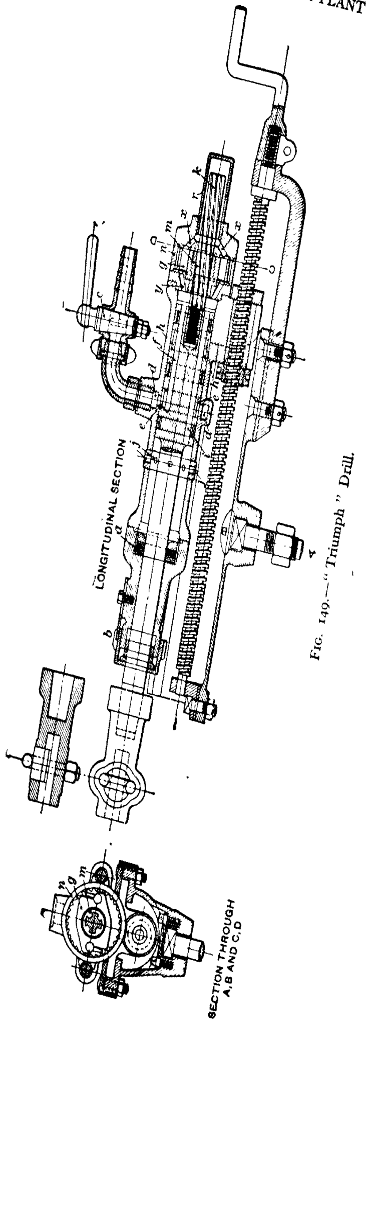


FIG. 149.—"Triumph" Drill.

Operation of Reciprocating Drills

Air Pressure. The evidence from recorded tests shows conclusively that low air pressure is uneconomical. The force of the blow and number of strokes per minute fall off, resulting in a marked decrease in footage of hole drilled. Though drilling in soft rock does not require so high an air pressure as for hard, the best results are obtained by pressures of 70–80 lbs. Practice has tended toward the use of higher pressures, up to 90 or 100 lbs.; but, though more work in some kinds of rock may be done by pressures above 80 lbs., the life of the drill is shortened and cost of repairs increased. The customary nearly uncushioned blow, under heavy air pressure on hard rock, is very destructive to the machine, and the bits are dulled sooner and are more apt to chip.

Several important series of tests on air consumption at different pressures were made on the Rand between 1904 and 1909. The last of these tests continued for nearly a year.* The rock was red granite, a large block of which was embedded in concrete. A quarry bar was used for mounting the drills. All holes were drilled vertically, with abundance of water. Two receivers holding 757 cu.ft. were employed, the pressure for each run being raised by the compressor to 80 lbs., after which the receivers were shut off. One drill at a time was operated, the run continuing until the receiver pressure dropped to 70 lbs. The drill was then stopped, and the depth and diameter of hole measured. Similar runs were made with pressures from 70 to 60, 60 to 50 lbs., etc. The receiver capacity, in terms of cu.ft. of free air, was calculated for each run and pressure, correction applied for temperature, and the air consumed based on the volume of free air at 70° F. and 24.8 ins. of barometer (corresponding to the Rand altitude of 5,000 ft.).

Eliminating results of runs indicating erratic behavior of

* *Mech. Engs. Assoc. of the Witwatersrand*, 1904 (Abs. in *Mines and Minerals*, Sept., 1904, p. 64); *Jour.* Transvaal Inst. Mech. Engs., Nov , 1907 and Feb., 1908; *Eng. & Min. Jour.*, Jan 21, and Feb. 18, 1911.

some of the drills, due to being in poor condition, a test of 13 $3\frac{1}{4}$-in. drills, with 3-in. bits, gave the following averages:

TABLE XXX

	Gage Pressure, Lbs				
	80-70	70-60	60-50	50-40	40-35
Linear inches drilled per min	1 3	1 1	1 0	0 6	0 5
Cu ft. free air per minute.	124	117	100	70.	60.
Cu.ft. free air per linear in. of hole. .	95 3	106 4	100	116 4	120.
Ditto per cu in. of hole.	13 3	14 8	13 8	15 0	16.6

Each run occupied about 6 mins. Some of the average results were not consistent, and individual figures, of course, showed still greater variations. These were due to lack of uniformity of the rock, differences in temper and sharpness of bits and the personal equation of the drill-runners, each of whom "was selected by the agent of the maker of the drill." The lengthy paper from which these data are taken includes many tables, thoroughly summarizing the work. Among other points, the importance of the question of air pressure is clearly demonstrated.

Consumption of Air. Due to the irregularity of the work of machine drilling, and the fact that a number of drills are generally operated by the same compressor plant, few figures are available as to the actual air consumption of a single machine. Average figures, however, are the only really useful ones. The duty is usually based on the consumption of free air per min., which depends on the size of drill, air pressure, character of rock, and the proportion of the total time actually occupied in drilling. The compressor capacity for one drill is evidently greater than the average required for a number of machines. With a large number, the delays to which each is subject, for setting up or shifting, changing bits, stoppages caused by the bit sticking in the hole, etc., make it improbable that all will be in simultaneous operation; hence, the average allowance of air for each may be reduced. Momentary or occasional peaks in the load on the

compressor, when an unusual number of drills are working simultaneously, may be disregarded; or at least need not be provided for by increasing the compressor capacity.

Rock-drills of different makers, even when of the same diameter of cylinder, vary in their consumption of air and reliable figures are not easily obtained. Table XXXI, showing the free air per minute required for one drill, is based on a comparison of the statements of several manufacturers, checked by recorded tests. It represents, within reasonable limits of error, actual practice for machines in good order. No allowance is made for the preventable loss of air in leaky pipes, nor for frictional loss of pressure in transmission (see Chap. XVI).

TABLE XXXI

CUBIC FEET OF FREE AIR PER MINUTE CONSUMED BY ONE DRILL AT SEA-LEVEL

Gage Pressure	Diameters of Drill Cylinder in Ins.											
	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{8}$	$3\frac{3}{16}$	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{5}{8}$	$4\frac{1}{4}$	5
60	58	63	70	82	90	97	100	105	114	118	135	155
70	62	72	80	92	104	112	115	118	130	135	152	174
80	70	80	88	103	115	125	130	135	142	153	173	205
90	78	87	95	115	128	137	141	148	165	173	194	222
100	85	96	108	126	140	151	155	161	176	184	210	250

When a number of drills are operated by the same plant, the compressor capacity for furnishing the total average quantity of free air required per minute, at sea-level, may be found approximately by the following table of multipliers:

TABLE XXXII

Number of drills	1	2	3	4	5	6	7	8	9	10
Multiplier	1	1.8	2.7	3.4	4.1	4.8	5.45	6.1	6.7	7.3
Number of drills.	11	12	15	20	25	30	35	40	50	60
Multiplier	7.8	8.4	10.3	12.8	15.1	17.3	19.7	22.0	26.5	30.5

To use the tables multiply the cubic feet of free air per minute consumed by one drill (Table XXXI), by the multiplier corresponding to the number of drills (Table XXXII).

To Find the Compressor Horse-Power required for any number of drills at any altitude. Example: 10 $2\frac{1}{4}$-in. drills, working at 12,000 ft. altitude, with air at 80 lbs. gage.

From Table XXXI, 1 drill at sea-level uses 80 cu.ft. free air, and 10 drills (Table XXXII), $80\times7.3=584$ cu.ft. At 12,000 ft., as the relative output for 80 lbs. (Table XIII, p. 194) is 0.65, the compressor capacity is $584\div0.65=900$ cu.ft. The volume of compressed air per I.H.P. at 80 lbs. and 12,000 ft. $=\frac{0.925+0.646}{2}=0.785$ cu.ft. In the equation $V'=\frac{VP}{P'}$, P, from Table XIII, $=9.34$; hence, $V'=\frac{900\times9.34}{80+9.34}=94$, and $\frac{94}{0.785}=$ 119.7 H.P.

The horse-power may also be found by using the ratio of compression directly, thus: cu.ft. at sea-level (see above) $=584$. Ratio of compression at 12,000 ft. $=\frac{80+9.34}{9.34}=9.56$, which (Table VI, p. 141) corresponds to 127 lbs. gage, and the H.P. required to compress 1 cu.ft. to this pressure $=0.204$. Hence, $584\times0.204=119.13$ H.P., which agrees closely with the result obtained by the first method.

The figures in Tables XXXI and XXXII are not exactly applicable to all cases. Modifying factors are:

(1) **Kind of Work.** The time required to set up the drill depends greatly on the shape of the working, whether a tunnel or drift, a shaft, stope, or open cut. If the floor and roof, or the side walls, of a mine opening are irregular or loose, much time may be lost in shifting the machine and setting it up, according as it is mounted on column or tripod.

(2) **Character of Rock** also influences the air consumption. In hard rock the advance in drilling is slower than in soft, so that the machine makes longer continuous runs. Less total time is occupied in shifting and setting up for drilling the successive holes of a round, and the consumption of air per unit

of time is therefore greater. Though this increase is partly offset by the fact that the bits are more quickly dulled in hard rock and must be changed at shorter intervals; still, in very hard ground the drills may be kept running with but few and short intermissions. In soft rock, though the actual speed of drilling is greater, there are apt to be more delays due to rifling of the hole and sticking or "fitchering" of the bit. On the whole, for hard rock it is advisable to provide a greater compressor capacity than is given in the tables. The compressor can then run at a slower speed, thus avoiding excessive heating of the air. The time actually occupied in drilling will vary for each machine from, say, 4 to 6 hours out of an 8-hour shift.

(3) **Physical Condition of the Drill** is important. The tabulated figures are for new machines, or those in good order. More air is consumed by old drills, with valves and pistons so worn that they do not fit closely. Even for drills in fair average condition, this is clearly shown by the fact that the exhaust, instead of being short and sharp, is nearly continuous. A large allowance must be made for old machines.

If definite values could be assigned to the above items, estimates of air consumption could be made for any given condition. Though this is manifestly impossible, a few averages for an entire shift's work have been recorded by Messrs. J. E. Bell and L. L. Summers. For a 3-in. drill, the volume of free air required per shift of 8-hours is as follows in Table XXXIII, the gage pressure being 100 lbs.:

TABLE XXXIII

Elevation.	Cubic Feet of Free Air.	
	Per Shift of 8 Hours	Per Minute.
Sea-level	25,000–42,000	52.1–87.5
5,000 ft	30,000–49,000	62.5–102.0
10,000 ft	35,000–60,000	73.0–125.0

These figures include all deductions for delays and stoppages. Taking the various allowances into account, and applying

them to Tables XXXI and XXXII, the following results of an elaborate test made at the Rose Deep Mine, Johannesburg, South Africa,* will be found in fairly close agreement with what precedes. The average number of drills (Ingersoll-Sergeant), of several sizes, operated during the 6-hour test, was calculated to be equivalent to 30.9 $3\frac{1}{4}$-in. drills. Average duty per drill, 4 ft. $5\frac{11}{16}$ ins. of hole per hour (diameter of hole not stated). Average air pressure, 69.83 lbs. Free air used per drill per minute, 81.08 cu.ft. It is fair to assume that most of the drills were more or less worn, or at least not in perfect condition. According to the tables, the average free-air consumption for 30.9 drills should have been 68 cu.ft. per min., or about 15% less than that shown by the test. This difference is due in part to the altitude above sea-level. The compressor horse-power per drill was 12.72; but as the work done during the 6-hour test was approximately equal to that usually accomplished in 8 hours of regular work, the actual horse-power per drill under normal conditions in this mine may be taken as $12.72 \times \frac{6}{8} = 9.54$. The air piping was known to be remarkably free from leaks.

Another test run, on 75 $3\frac{1}{8}$-in. drills was made at the Champion Iron Mine, Michigan.† At 78 lbs. normal gage pressure the average consumption for the day shift for 1 month was 67.1 cu.ft. of free air per min. The pressure usually dropped considerably, however, when work was in active progress. According to the tables, 75 drills should use about 58.5 cu.ft. of free air per min., or 13% less than shown by the test.

A large part of the compressed air used in the mines of the central Rand is now bought from the Rand Mines Power Supply Co., on meter measurement at a pressure of about 100 lbs. The following notes are from a paper by E. G. Izod and E. J. Laschinger (*Trans.* So. Af. Inst. M. E., Oct., 1913). The unit of measurement is 440 cu.ft. at 12.1 lbs. (Rand atmospheric pressure), weighed at 60° F.; it represents 641 watt-hours, which is the energy developed by the isothermal expansion of 440 cu.ft. of air, from 100 lbs. to 0 lb. gage.

* L. I. Seymour, *South African Association of Engineers*, 1898.

† *Eng. and Min. Jour.*, May 18, 1905, p. 937.

A $3\frac{1}{4}$-in. rock-drill in stoping uses an average of 80 air-power units per 8-hour shift, equivalent to 35,200 cu.ft. free Rand air; on development work, about 70,000 cu.ft. A useful curve diagram for checking air consumption is obtained by platting the average air units per drill shift as ordinates, and the ratio of development shifts to total shifts as abscissæ. By attention to this, one large group of mines reduced its air consumption per drill shift from 163 to 138 units. Paying for air by meter leads to more care in its use.

At the Village Main Reef Mine extensive tests were made in 1913, as summarized in Table XXXIV. They show the importance of a close fit of the piston in the cylinder.

TABLE XXXIV

TESTS ON A 3.183-INCH INGERSOLL DRILL

Gage Pressure.	70 Lbs.		90 Lbs	
Piston diam., ins	3 177	3 157	3 177	3 157
Difference between cyl. bore and piston diam, ins.	0 006	0 026	0 006	0 026
Strokes per min..	236	235	250	262
Lbs. free air per min. .	6 9	9 4	10 2	12 8
Lbs. air per double stroke	0 0292	0 04	0 0408	0.049
Cu.ft. free air per min. at 68° F and 12.2 lbs. absolute pressure	112	152	165	207

Efficiency. Though it is well-known that compressed-air drills are uneconomical in consumption of power, it is difficult to reach definite conclusions as to their efficiency. The useful work in the ordinary mechanical sense, done by a drill in making a hole of given depth and diameter in a rock of given hardness, toughness, and general physical character cannot be determined absolutely. All that is known is that the drill requires a certain volume of air per minute, which has been furnished by the expenditure of a certain average horse-power at the compressor. Comparative figures of work done, in terms of speed of drilling per cubic foot of free air consumed, are useful as far as they go, and are the only practical basis for estimating efficiency. But

the results obtained do not accurately represent the efficiency of machine drills as compared with other air or steam engines.

In their operation, rock-drills differ greatly from other compressed-air machines, because the personal element of the skill and experience of the drill-runner exerts so important an influence upon the amount of work accomplished, and because the rate of drilling is so greatly modified by the physical and mineralogical character of the rock, together with the purely adventitious occurrence of cracks, slips, and fissures. A skillful drill-runner will inevitably do more work per shift than an inexperienced man, and will make a faster advance in rock with which he is familiar than in rock that is new to him.

Therefore, though mechanical efficiency is the basis upon which machines in general are compared, in the case of compressed-air drills it is not the only consideration, nor is it the most important. Their efficiency is subordinate to the attributes of strength, simplicity, portability, durability, facility with which repairs may be made and capacity for work in terms of depth of hole drilled per unit of time. They must withstand hard and often unintelligent usage. The strong point of compressed-air drills is their ready applicability in their special field of work. In possessing a cylinder, piston, and valve, the drill roughly resembles a steam engine, but there the likeness ceases. Severe shock and vibration are essential accompaniments of its work. No fly-wheel is admissible, or other means of storing up and equalizing the power. The service demanded is therefore totally different from that performed by ordinary engines.

The low mechanical efficiency of the rock-drill is due mainly to the fact that air is admitted to the cylinder practically throughout full stroke. Hence, the valve motion resembles that of many simple, direct-acting pumps. Expansive use of the air to any extent is not practicable, both because of the undesirability of introducing complexity of mechanism in machines subjected to rough usage and the difficulty of adapting cutoff gear to the variable length of stroke required. The drill

cannot be kept always at full stroke; while in operation it is often necessary to feed the machine so far forward that the length of stroke is no more than 1 in., and the valve must still be able to cause a sharp, quick reversal of the stroke. The useful work is done on the forward stroke, in striking the blow. If the valve be thrown too soon, the stroke of the piston will be shortened; if too late, the piston will strike the cylinder head. Rock-drills, therefore, cannot attain the economy resulting in other air motors from using the air expansively. Incidentally, using air at full stroke is of some advantage, because exhausting at high pressure in a measure prevents troublesome accumulation of ice, in case the air is moist. Freezing, if any, is at least confined to the outer portion of the exhaust port, whence it is easily removed.

In dry, dusty mines, tappet-valve drills give good service. When a drill is not in use, and disconnected from the air hose, dust and grit may enter through the ports, passing thence into the valve chest and cylinder on resuming drilling. The wear and consequent looseness in the fit of the moving parts thus caused is apt to have a more unfavorable effect on the operation of the spool than the tappet valve. Leakage of air past the valve or piston prevents proper action of the auxiliary ports, not only producing irregularity in reversal and shortening of the stroke, but diminishing the drill's efficiency. It is true that the tappet valve involves the use of one extra part and, in case of the pivoted three-arm tappet, breakage is not infrequent. But while the spool valve is strong and reliable, its maintenance cost in dusty mines is higher than that of the tappet-valve motion.

The maximum force of blow is attained by drills working without cutoff on the forward stroke, and the best drills are thus designed. The valve is not reversed until the blow is delivered, and the exhaust is free, with but little back-pressure on the piston. Cushioning was formerly a feature of some drills, with the idea of reducing shock, but it is now recognized that an uncushioned blow gives greater efficiency. A drill so designed may strike too heavy a blow in very hard rock, the remedy

being to feed the drill-head down, so as to work with shorter stroke.

On the back stroke cushioning is desirable, to ease the reversal and prevent injury due to the piston striking the cylinder head. The back-stroke cushion is produced by cutting off the exhaust before the end of the stroke. Only enough power needs to be developed on this stroke to overcome the resistance due to the weight of the moving parts, and the frequent tendency for the bit to stick in the hole.

In ordinary machine drills, the piston speed should not be too great—say, not much over 350–375 strokes per minute. Relative speeds of stroke do not constitute a proper basis for the comparison of efficiencies. To give effect to the blow, the weight of the moving parts must be relatively great, and very high speed is attended by excessive wear and breakage. These conclusions do not apply to drills of the hammer type (Chap. XXI), which strike a light blow at a high speed; the weight of the moving parts is small.

Drill Repairs. In choosing a drill the question of repairs is of great importance. But little useful information concerning this point is available; such data could be obtained only by operating drills of several different kinds under the same conditions, and for a considerable period of time. Such opportunities are rare. It is generally inadvisable to use different makes or more than two sizes of drill in the same mine, since this requires the keeping of duplicate spare parts for each.

The item of repairs depends largely upon the experience and character of the drill runner. A careful man treats his machine with intelligent consideration. He will set it up properly, to reduce the risk of getting out of alignment as the hole is deepened; and, if the bit should stick ("fitcher"), he keeps his temper and refrains from striking unnecessarily heavy blows on the drill head or chuck. A fitchered bit may often be loosened by slacking the clamp bolts, thus allowing the machine slightly to shift its position. The serious abuse to which machine drills are frequently subjected may be reduced by efficient supervision on the part of foremen and shift-bosses.

In every machine there should be as few moving parts as possible. But the work of a rock-drill is severe, and it is often operated by incompetent men. Even when run with care, wear is rapid and breakages are frequent. The maker therefore has the problem of producing a drill comprising as few parts as practicable, and designing it so that the parts especially liable to wear or breakage may be replaced readily, cheaply and without the necessity of discarding a larger piece, with which the broken part may be connected. New cylinders may be bored out to fit worn pistons, or new pistons fitted to old cylinders.

When new, the piston should have the closest possible working fit. The difference between the cylinder bore and piston diameter should not exceed $\frac{1}{64}$-in. Modern drill cylinders are counterbored at each end, to facilitate reboring. The difference in diameter between a spool-valve and the bore of its chest should not exceed $\frac{1}{100}$-in. Chuck and chuck keys must be in good order, and the bit when set must be in accurate alignment with the piston. A well-equipped repair shop lengthens the life of machine drills. Drill runners should not be encouraged to tinker their machines underground. If repairs or adjustment be necessary, the drill should be sent at once to the shop.*

Repair Costs. While no generalizations are possible, on account of the extremely variable service of rock-drills, Tables XXXV and XXXVI, C. K. Hitchcock, Jr. (*S. of M. Quarterly*, 1914), of repairs costs at a Michigan copper mine, will be found significant and useful.

The records of cost of labor and material (not including mountings nor connections) were kept by the month, the cost at the end of each month being platted on a chart against the number of days, of two shifts, that the drill had worked since being put into commission. From the chart the costs for each period of 25 days were scaled, as recorded in the tables. Most of the smaller repair parts were made in the mine shops, the larger parts being bought from the makers. Table XXXV

* For a detailed discussion of maintenance and repairs see Chap. VIII, "Rock Drills," by E. M. Weston. See also *Min. and Sci. Press.*, 1905, Nov. 4, p. 308; Nov. 11, p. 329.

shows that the average costs between 100 and 150 days of service were about $7.00 for each 25 days.

Table XXXV
Repair Costs of 15 New $2\frac{3}{4}$-Inch Drills

Drill	Days in Use, 2 Shifts per Day						
	25	50	75	100	125	150	175
1	$ 0 60	$ 1 25	$ 1 50	$ 2 40	$ 3 00	$ 7 05	$10 80
2	3 75	7 50	7 90	8 75	12 00	16 25	21 80
3	3 30	10 00	11 10	13 50	20 30	40 75	40 91
4	90	1 25	4 55	5 30	28 39	28 39	28 39
5	15	40	1 50	4 50	5 80	6 40	9 20
6	3 80	7 40	8 00	8 08	8 08	12 15	
7	20	50	1 10	4 75	8 00	14 95	16 20
8	55	1 05	5 60	5 91	7 80	10 57	10 70
9	4 00	6 20	8 70	12 30	41 20	54 59	
10	70	70	3 50	4 70	8 29	8 45	
11	60	1 00	1 50	21 00	41 00	42 10	43 36
12	55	1 00	2 20	9 10	12 00	12 30	21 30
13	85	1 20	4 40	7 70	8 70	10 25	18 14
14	3 65	4 10	4 85	5 05	8 50	50 50	52 01
15	2 55	4 35	4 65	5 20	11 20	14 60	14 75
Totals..	$26 15	$47 90	$78 05	$118 24	$224 26	$329 30	
Averages	1 74	3 19	4 74	7 88	14 95	21 95	

Table XXXVI indicates a repair cost of about $5.00 per 25 days. In the upkeep of machine drills, both economy and efficiency can be improved by discarding drills having abnormally poor records, or by overhauling them so thoroughly as to cut down cost of maintenance.

At a large group of mines on the Rand, the complete repair cost of standard piston drills (including air hose) averaged $42.00 per 52 shifts. On introducing a contract system of drill repairs, the cost was reduced to $27.00, and it is thought it might be further reduced to $21.00 or $23.00.

Records of Work. Referring to the preceding discussion of the operation of machine drills, if approximately complete records of work could be secured, useful tabulations could be made of the comparative speeds of drilling in different rocks

Table XXXVI

Repair Costs of Six 3⅛-Inch Drills

Drill.	Number of Days in Use					
	25	50	75	100	125	150
A	$ 2 00	$ 2 50	$ 4 00	$ 5 40	$ 8 00	$ 19.00
B	3 00	11 30	19 10	22 30	27 00	34 00
C	6 10	14 20	18 20	19 60	24 50	28 60
D	4 20	5 20	6 20	9 20	18 10	18 20
E	28 00	33 30	36 10	37 00	38 10	41 50
F	50	4 30	4 90	11 00	12 80	14 00
Totals.	$ 43 80	$ 70 80	$ 88 50	$104 50	$128 50	$155.30
Averages	7 30	11 80	14 75	17 42	21 42	25 88

Drill	Number of Days in Use					
	175	200	225	250	275	300
A	$ 22 00	$ 22 60	$ 40 00	$ 64 20	$84 80	$ 88 10
B	40 20	41 30	42 20	46 70	53 20	59 60
C	28 90	28 90	32 60	33 40	46 30	49 20
D	18 30	21 50	22 10	46 00	46 00	47 60
E	44 70	45 90	46 50	46 90	48 20	49 50
F	17 80	18 20	18 20	35 00	55 20	59 30
Totals.	$171 90	$178 40	$201 60	$272 20	$333 70	$353 30
Averages	28 65	29 73	33 60	45 37	55 62	58 88

Drill	Number of Days in Use.					
	325	350	375	400	425	450
A	$ 97 00	$ 99 50	$101 50	$102 00	$115 20	
B	66 00	66 80	67 00	78.80	79 50	
C	50 20	48 80	54 50	61 00	75 00	$76.70
D	48 20	53 70	54 60	57.10	60 20	62.70
E	72 00	90 50	111 20	112 00	113 80	113 80
F	60 20	60 70	63 00	63 70	64 00	70 00
Totals. . . .	$393 60	$420 00	$451 80	$474 60	$508 00	
Averages. . .	65 60	70 00	75 30	79 10	84.67	

and ores. Having such data, it might even be possible to designate the kinds of service for which the different makes and types of drill are best adapted. But, rock and ore characteristics, and other local conditions, vary so greatly that detailed comparisons are of doubtful value. The results of " test runs " are sometimes cited to show that one machine is better than another, in the sense that it can drill faster or that it uses less air for the same footage. By operating drills of different type side by side, in the same rock and with the same air pressure, some approach to an accurate comparison would be possible; but, even then, it is clear that the physical condition of the competing drills, and the " personal equations " of the respective drill runners, affect the rate of work and cost per foot of hole. Moreover, as test runs are made under the stimulus of rivalry (and perhaps with a " bonus " to the winner), they do not constitute a useful basis for ascertaining the relative merits of different drills, nor the footage that could reasonably be expected in ordinary daily work. For further elucidation of these considerations the reader is referred to the voluminous records of the elaborate drill tests made on the Rand (see footnote, p. 295).

In recording the work of machine drills, the important items are:

(1) Kind of rock or ore.
(2) Type and size of drill.
(3) Air pressure at the drill.
(4) Diameter of hole.
(5) Total elapsed time for a hole of given depth.
(6) Measured volumes of free and compressed air consumed for the depth drilled.
(7) Time occupied in setting up, changing bits, and delays due to breakage or derangement of the drill, or to " fitchering " of bits.
(8) Net drilling time per hole and per foot or inch of hole, resulting in the number of inches per minute (or feet per hour) drilled while the machine is in actual operation.

Examples of Drilling Speeds. Table XXXVII gives figures based on a number of recorded runs.

TABLE XXXVII

Kind of Work.	Rock or Ore.	Size of Drill, Ins.	Air Pressure, Lbs.	Aver. Depth of Hole, Ft	Inches of Hole per Minute. Total Time.	Net Time.
Crosscut, 5×7 ft	Hard quartzite and limestone.	3	60	3	0 43	
Crosscut, 5×7 ft	Hard quartzite and limestone.	3	60	$3\frac{3}{4}$	0 40	
Drift.	Quartzite	$3\frac{1}{8}$	80	$5\frac{1}{2}$	1 29	1 55
Tunnel, 7×7 ft.	Basalt.	3			0 72	
Stoping	Amygdaloid copper rock.	$3\frac{1}{8}$	63	7	0 93	
Stoping	Amygdaloid copper rock	$3\frac{1}{8}$	63	6	1 60	2 30
Drift.	Amygdaloid copper rock.	$3\frac{1}{8}$	65	3	1 26	2 70
Stoping and drifting	Hard limestone	$2\frac{1}{4}$ / $2\frac{3}{4}$	80	7	1 00	
Stoping	Hard limestone	$2\frac{1}{4}$	75	$6\frac{3}{4}$		2 46
Stoping	Magnetic iron ore.	3	80	$6\frac{1}{2}$	1 31	
Stoping	Rather soft porphyry, etc.	$2\frac{3}{4}$	75	$4\frac{2}{3}$	1 38	
Stoping	Very hard quartzite	$2\frac{3}{4}$	75	$3\frac{1}{8}$	0 54	
Crosscut	Quartz and hard slate	$2\frac{3}{4}$	73		1 50	
Stoping	Hard quartz.	$2\frac{1}{4}$	73		1 20	
Stoping	Hard hematite	3	74	7	1 28	
Stoping	Quartzose	$2\frac{3}{4}$		$5\frac{1}{3}$	1 51	2 25
Drift, 10×10 ft	Hard limestone	$3\frac{1}{4}$		4	1 35	
Stoping	Pyritic ore	$3\frac{1}{8}$		4	1 20	
Drift, 4.5×6 ft	Limestone	$2\frac{3}{4}$	105	$8\frac{1}{3}$	2 62	
Stoping	Amygdaloid copper rock.	3 & $3\frac{1}{8}$		7	1 67	2.45
Drift	Amygdaloid copper rock.	3 & $3\frac{1}{8}$	. .	$5\frac{2}{3}$	1 46	2.34
Drift	Amygdaloid copper rock.	3 & $3\frac{1}{8}$		$5\frac{2}{3}$	1 61	1 97
Stoping	Hard phonolite breccia	$2\frac{1}{4}$		$2\frac{1}{3}$	0 89	0.95
Drift	Hard phonolite breccia	$2\frac{1}{4}$	.	$4\frac{1}{2}$	2 60	3 38
Crosscut, 9×9 ft	Quartzite	$2\frac{3}{4}$	.	$3\frac{2}{3}$	1 17	2 43

Conclusions. The average speed of drilling based on column 6 of Table XXXVII is 6.4 ft. per hour. In general the duty of a standard 3-in. drill, in rock or ore of average hardness,

ranges from 40–50 ft. per 8-hour shift, including all ordinary delays for setting up and changing bits. For very hard, tough ground, the speed is often lower, while much more than 50 ft. per shift may be made when the conditions are favorable, and also in drilling deep holes, for which fewer set-ups are required. The cost per foot of hole is extremely variable, ranging from say 8 cents in easy ground, and where wages are low, up to 25 cents under adverse conditions.

For moderately soft ground, not requiring holes of large diameter to contain the necessary quantity of powder, the smaller sizes of machine drill (2–2½ in.) are usually preferable. Their first cost and air consumption are less than for large drills, and they may be operated by one man. Small machines are especially useful for stoping in thin veins. For hard ground, and as a rule in shaft sinking, tunnelling, cross-cutting and similar work, the 2¾, 3, and 3⅛-in. sizes are best. For deep holes in open-cut work, still heavier drills are often necessary—up to 3½ in., or even larger.

Hammer drills (Chap. XXI) are now successful competitors of reciprocating drills for nearly all kinds of rock excavation, including many of the operations of mining.

CHAPTER XXI

HAMMER DRILLS

THE principle of the hammer drill was first applied in pneumatic riveters, and tools for chipping, rough chiseling and miscellaneous machine shop work. Their earliest employment in mines was for cutting hitches for timbers, blockholing, and other shallow drilling. In recent years they have rapidly grown in favor. They have largely displaced reciprocating machines for stoping and similar work, and are often used for sinking shafts and winzes. In other words, they are best applied to drilling holes directed steeply upward or downward. For tunnelling, drifting, cross-cutting, etc., reciprocating drills are still in general use, though hammer drills are employed to some extent in these operations also.

Classification. The author proposes the following classification, as being more useful than one based upon the type of valve-motion:

A. Machines designed for tripod or column mounting; usually of the larger sizes, comparable to standard drills of the reciprocating type.

B. Small machines with either a cross or D-shaped handle; chiefly for drilling holes directed steeply downward, the weight of the drill resting on the bit itself.

C. Machines with an automatic air-feed standard; primarily for overhead stoping, though they may also be mounted on a column, as for breast work.

General Construction. In the hammer drill, the bit does not reciprocate; its shank projects into the forward end of the cylinder and is struck a rapid succession of blows by the piston, which acts as a hammer. The cutting edge of the bit is in constant contact with the bottom of the hole, except during the

slight rebound caused by each blow of the piston. Some of these machines, like the Hardsocg, are valveless, the functions of the valve being performed by the reciprocations of the piston. Others, like the Leyner-Ingersoll, Sullivan, and Climax, have air-thrown valves. At first, no attempt was made to introduce automatic rotation of the bit; the operator simply turned the drill back and forth on its axis, by means of the handle. At the present time, most hammer drills are provided with rifle-bar rotations, similar to that of the reciprocating machines, or some modification of that device. The bit shank is octagonal, generally fitting loosely in a corresponding chuck socket. To keep the hole round and reduce the chances of rifling, the bit is commonly of the star or rosette shape, with 6 (sometimes 8) radial cutting edges. The cutting edges are thus so close together that even if several successive blows are struck in the same angular position of the bit, the ridges of rock between the edges are broken away.

As the bit does not reciprocate, it is evident that, for down holes more than a few inches deep, some automatic means must be provided for removing the drill dust or sludge and keeping the bottom of the hole clean; otherwise much of the useful effect of the hammer blows would be lost. To accomplish this, a hollow bit is generally used, a small hole being bored longitudinally through its center. By injecting a jet of water, the drillings are displaced and the bit is kept cool. The same result is attained by a jet of compressed air, which produces a low temperature on expanding. As the speed of stroke is great, the cooling of the bit in dry holes is important. The water jet is usually applied as described under the Sullivan "Liteweight" drill, Chap. XX, p. 278 (see also the "Leyner-Ingersoll drill, below). When an air jet is used, it is delivered through a similar axial tube in the drill, the rear end of the tube being connected with the air-feed pipe; or a small quantity of compressed air may be led directly to the hollow bit from the valve chest.

The small hand hammer drills (class B), used chiefly for down holes, are fed simply by keeping the bit pressed firmly

against the bottom of the hole. Usually the automatic feed, for stoping drills (class C), consists of a light telescopic standard, attached to the drill head. It is supplied with compressed air, which keeps the drill fed up to its work as the hole is deepened. Incidentally this device furnishes an air cushion, relieving the operator handling the drill from much of the annoyance caused by vibration. For breast-stoping, drifting, and other horizontal work, these machines may be mounted on a light column.

Examples of each class of hammer drills are given below.

CLASS A. LARGE HAMMER DRILLS

Leyner-Ingersoll Water Drills, Nos. 18 and 26 (Figs. 150, 151), are provided with the "Butterfly" valve, for an illustrated description of which see Ingersoll-Rand "Butterfly" reciprocating drill (Chap. XX, p. 283).

Rotation of the bit is effected as follows (see also Fig. 152): The rifle-bar 1, which is provided with a ratchet and three pawls, engages with a rifle-nut, screwed into the hollow rear end of the hammer 2. This causes rotation of the hammer on each back stroke. The forward, smaller end 3 of the hammer is fluted, and engages with an internally fluted bronze nut in the rear end of the chuck. Thus the chuck, holding the bit, is caused to rotate with the hammer.

The drill bit is hollow, for the passage of the water into the drill hole. The bit for the No. 18 drill has two lugs by which it is locked in the chuck. Fig. 153 shows the bit shanks of both drills. Another model of the No. 18 drill has an "anvil block," inserted between the hammer and bit. The steel has no lugs, but fits in a bushing screwed into the chuck, and the hammer strikes the anvil block, instead of the bit. In hard ground, which does not "ravel" (run into the hole on withdrawing the bit), this design works well. By omitting the lugs, there is a saving in blacksmithing, and, as the drill-runner cannot back the machine out of the hole with full air pressure on, breakage of parts is reduced. The smaller drill, No. 26 (Fig. 152), has a different chuck, designed for bits formed with

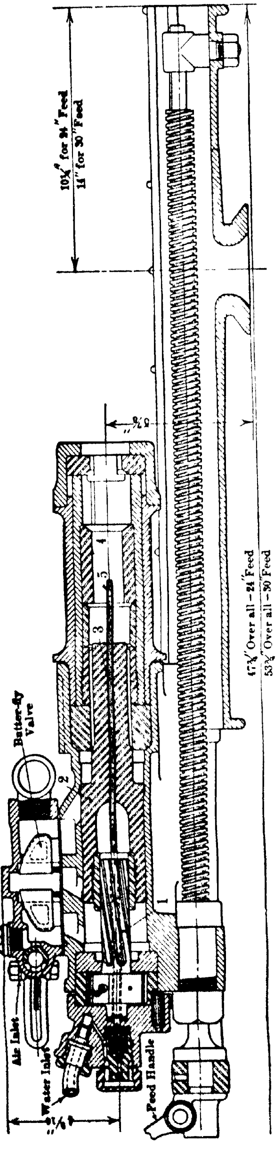


FIG. 150.—Leyner-Ingersoll Water Drill, No. 18.

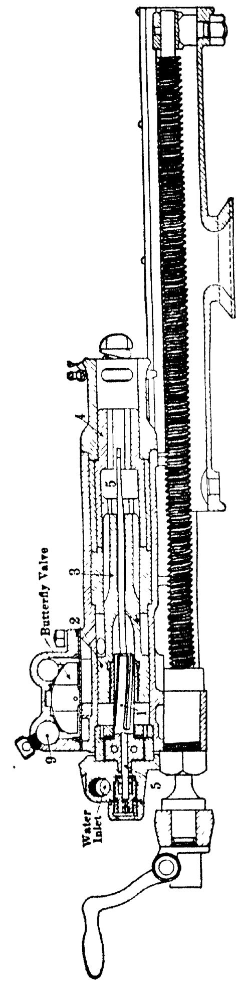


FIG. 151.—Leyner-Ingersoll Water Drill, No. 26.

a collar and hexagon shank. To retain the bit in the chuck, a pin 7 is dropped into a hole in the front head. The weight of the drill only is 95 lbs.

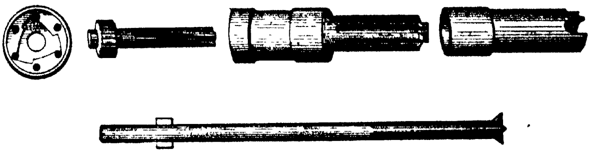


FIG. 152.—Rotation Device of Leyner-Ingersoll Drill, No. 18.*

The water supply is furnished under pressure from an 18-gal. steel tank, weighing 70 lbs., accompanying the drill and connected to it by a length of hose, as shown in Fig. 136, Chap. XX.

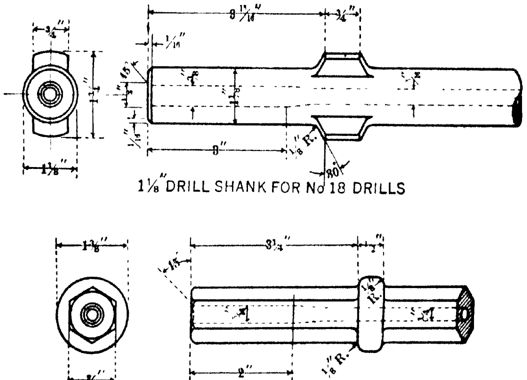


FIG. 153.—Details of Bit Shanks.

Another hose conveys compressed air from the main to the tank. Water is thus forced from the tank through the water tube 5 (Fig. 150), which passes through the rifle-bar and hammer,

* The smaller details of design of the present rotation mechanism differ somewhat from those shown in this cut, but the principle is unchanged.

in the axis of the machine, and is delivered into the hollow bit. Air is mixed with the water, as the hole is thus cleaned more effectually than by water alone. An air jet, without water, makes too much dust.* The air comes from the drill when in operation, not from the tank.

Lubrication. In the No. 18 drill, an oil chamber is cast under the cylinder bore. It is filled through a plugged opening on top of the cylinder. There is a small port between the oil chamber and cylinder, so that, as the piston travels over this oil port, the oil in the chamber is alternately under live air and exhaust pressure. A small quantity of oil is thus fed to the cylinder at each stroke. The No. 26 drill has an oil chamber 9 (Fig. 151), on top of the cylinder, provided with a patented oiler feeding into the valve chest.

The latest (1918) Leyner-Ingersoll drills are the Nos. 148 and 248 (Fig. 153*a*). Though there are slight modifications in construction, the rotation and water features are as in the No. 18 drill. These machines are identical, except that No. 148 has a light shell of 24-in. feed only, while No. 248 may be had with either 24 or 30-in. feed. They are valveless as to admission and exhaust, the hammer being of the differential type; but there is a valve in the cylinder to eliminate back pressure and permit an uncushioned blow. Lubrication is by the automatic "Heart-beat" lubricator, connected by a small port with the rear end of the cylinder, and operated by the air pulsations. Its oil-carrying "cartridge" is recharged as necessary. Weight of drills: No. 148, 148 lbs.; No. 248, 156 lbs.

Sullivan "DR-6" Drill (Fig. 154) is mounted on a column or tripod, and is designed primarily for drifting and tunnelling. Cylinder diameter, 2¼ ins.; net weight, 148 lbs.; diameter of drill steel, 1¼ in.

* In late years the dust question has received much attention. Though dust from shales, coal and other non-siliceous material is comparatively harmless, that coming from siliceous rock or ore is distinctly dangerous, causing "silicosis" or miner's consumption. For a discussion of this important subject see Mining Engineer's Handbook (Peele, 1918), pp. 1379, 1402.

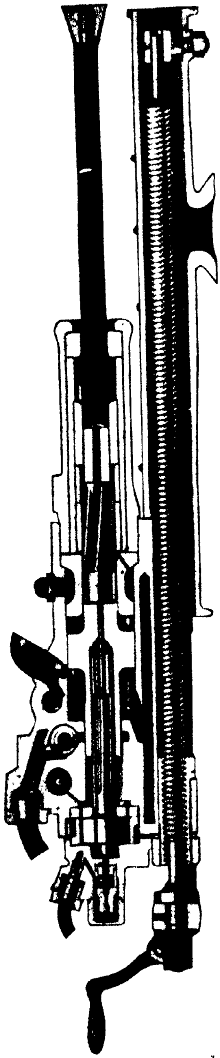


FIG. 153*a*.—Leyner-Ingersoll Water Drill, Nos 148 and 248.

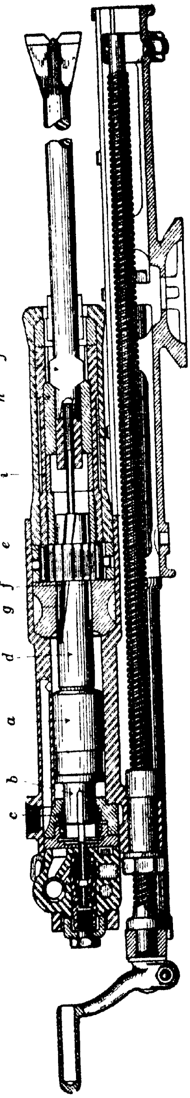


FIG. 154.—Sullivan "DR-6" Drill, with Water Attachment.

The hammer *a* has a short tail piece *b*, which runs inside of a hollow shell valve *c*, having end seats and placed in the rear of the cylinder. The rotation ratchet *e* is set in the forward end of the drill. A long front extension *d* of the hammer, which serves as a rifle-bar, passes through the ratchet sleeve, and strikes directly on the drill shank. Straight and spiral grooves, *f* and *g*, are cut in the rifled striking end *d* of the hammer; the straight grooves engage with guides in the retaining bushing *i*, which incloses the drill-shank bushing *h*, and the spiral grooves engage with projections in the ratchet sleeve. There is no rotation of the bit on the forward stroke. On the back stroke the ratchet causes the hammer to ride up on its spiral fluting, thus rotating it; and since the hammer, while reciprocating, engages with bushing *i* by means of the straight grooves, the chuck bushing *h* and the bit are also rotated. Like the Leyner-Ingersoll bit, previously described, the bit shank has two lugs *j*, fitting in the chuck, to provide the grip for transmitting the rotation.

This machine has a water attachment like that used in the "Liteweight" drill (Chap. XX). It comprises a combined water and air jet, a single throttle valve controlling both air and water. Air alone may be used for shallow holes.

Stephens' "Climax Imperial" Hammer Drill, made at Carn Brea, Cornwall, has a $1\frac{3}{4}$-in. cylinder, weighs 75 lbs., and is mounted on a light column or bar. In several features of its design this machine differs greatly from American drills.

The valve motion (Fig. 155) resembles that of the Climax reciprocating drill (Chap. XX). Air enters by the combined air and water tap (detailed section) on the side of the valve chest; thence passing by the annular recess *b* in the piston valve *a*, through *c* and *c'*, to the main cylinder ports *h*,*h'*. The recesses *d*,*d'*, in the valve, communicate with the main exhaust (not shown). Air is constantly admitted to both ends of the chest by a small groove *t*, the valve being thrown by exhausting through the much larger auxiliary ports *e* and *f*.* Ports *f* are alternately brought into communication with the annular recess

* The device is similar to that in the "Sergeant" drill, Fig. 131, Chap. XX.

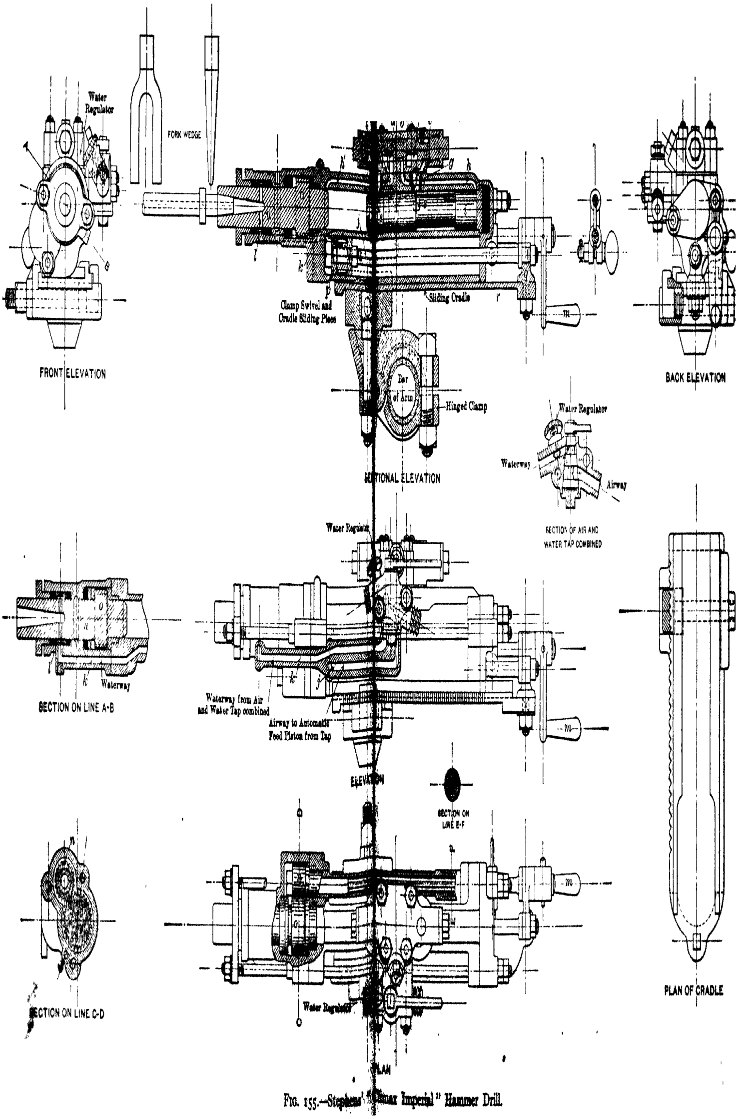


FIG. 155.—Stephens' "Climax Imperial" Hammer Drill.

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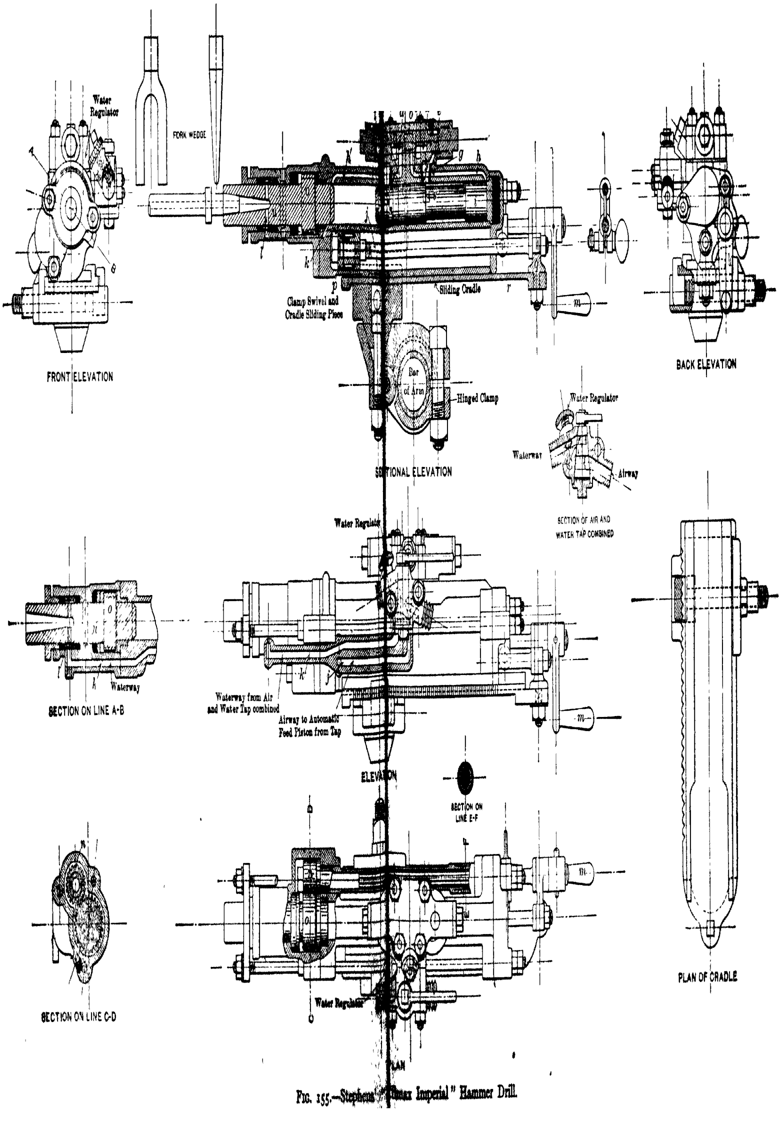


FIG. 155.—Stephens' "Climax Imperial" Hammer Drill.

To face page 318.

h, thus releasing the air, by way of ports s and s', to the main exhaust. The ports f are lined with hollow, conical plugs g, of composition metal, shaped below to the curve of the hammer; to prevent leakage of air, they are kept in close contact with the hammer by the pressure of the valve-chest, when bolted in place. When the plugs wear too loose, a thin washer is inserted above them.

The water for the drill hole is best supplied by gravity, under a pressure of say 15 lbs. It enters the combined air and water tap, or throttle, through the passage k, to the transverse port l in the anvil block u which serves also as the drill holder, or chuck. Thence the water passes to the hollow bit (see the elevation and the "section on line AB"). The drill may also be used for "dry" holes, the dust being allayed by an external spray from the throttle.*

The machine has an automatic air-feed cylinder. A small piston p, with its rod, is rigidly bolted to the lug q, on the cradle r. Air from the throttle passes through passage j to the feed cylinder, forcing the drill head forward on the cradle and keeping the bit pressed against the bottom of the hole. After the machine has been fed forward 14 ins. (the working length of feed), the air is shut off and the transverse bolt, shown in the plan of the cradle, is slacked. The operator then slides the cradle forward on its support under the machine, tightens up the bolt on the serrated edge of the cradle, and proceeds with the drilling. Thus, a total feed of twice the length of the cradle—or about 28 ins.— is obtained without putting in a longer bit.

Rotation of the bit is effected by hand. The bit is held by friction in the conical socket of the chuck. Gear teeth are cut on the periphery of an enlarged part o of the chuck, engaging with which is a smaller gear n (see general plan and the "section on line CD"), keyed on a spindle passing to the rear of the machine and rotated by the handle m.

Excellent drilling records have been made by this machine, both in England and South Africa.

* This "dust allayer" is described in Chap. XX, under Climax Drill.

CLASS B. SMALL HAMMER DRILLS, WITH HANDLE

Hardsocg Wonder Drill* (Figs. 156–7) is a valveless machine, made in two types. Fig. 156 shows the older form, with D-handle for hand rotation. There are 4 sizes, weighing 12, 17, 20 and 30 lbs., all using hollow steel and employed chiefly for down holes.

Air is admitted at the nipple 2 (to which is attached the throttle and hose) and, entering the annular recess 3, acts constantly on the shoulder 13 of the hammer. On beginning the forward stroke, the ports 5 and 6, through the head of the hammer, are opposite recess 3, and admit live air into and behind the hollow hammer. Since the area thus presented to air pressure is much greater than the area of shoulder 13, the

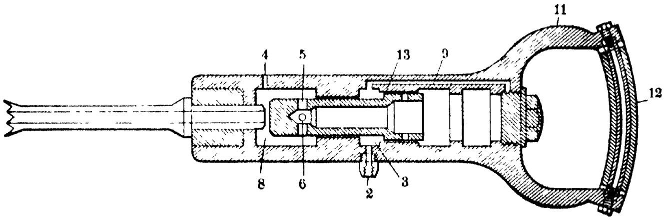


FIG. 156.—Hardsocg Wonder Drill, with D-Handle.

hammer is driven forward. Just before the hammer strikes the bit, ports 5, 6 reach the annular recess 8; through this recess, part of the air in and behind the hammer which has caused the forward stroke, goes to the exhaust port 4, and part through the hollow bit to the bottom of the drill hole. The exhaust having taken place, the back stroke is made by the constant pressure of the inlet air on the annular shoulder 13 of the hammer.

This drill can be converted into a stoping or drifting machine by removing the large plug and handle 12, and screwing into the rear head of the cylinder an air-feed standard, somewhat similar to that shown by Figs. 178, 181 and 183. For making breast holes, the air-feed machine is mounted on a light column. Weights, unmounted, 35 and 87 lbs.

* This drill was one of the earliest of the Hammer drills.

Fig. 157 shows a recent form of the Hardsocg drill, with automatic rotation: *a* is the ratchet box, with spring-controlled pawls; *b* the rifle-bar, engaging the rifle-nut *c* in the rear end of the hollow hammer *d*. The hammer is thus rotated like the piston of a reciprocating drill (Chap. XX). The forward

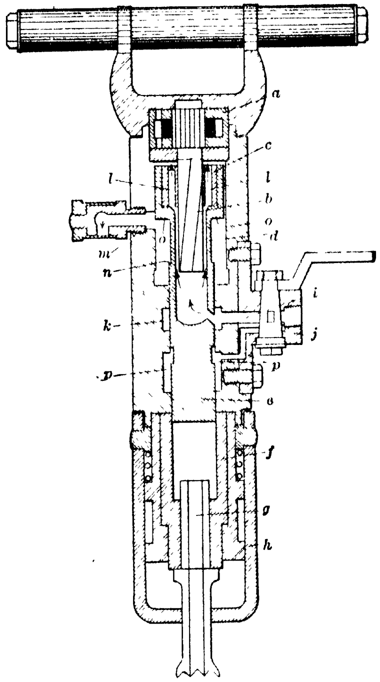


FIG. 157.—Hardsocg Drill, with Automatic Rotation.

end *e* of the hammer has a square cross-section, reciprocating in the square bushing *f*, which holds the octagon shank *g* of the bit. Thus the rotation of the hammer is communicated to the bit. A yoke-shaped bit retainer *h*, somewhat resembling that in Fig. 158, holds the bit in its socket.

Live air enters port *i*, from the throttle *j* to the annular recess *k*, and, as shown by the arrows, passes alongside the

rifle-bar, through passages *l,l*, in the rifle-nut, to the rear of the hammer. On completing the forward stroke, the air in the rear end of the cylinder passes to the exhaust port *m*. When the hammer is in this position, the small shoulder *n* is opposite the annular recess *k*, thus admitting air to act on the large shoulder *o* of the hammer, and causing the back stroke. At the end of the forward stroke, some of the exhaust air passes down alongside the square head of the hammer (which makes a loose fit in bushing *f*), and goes through the hollow steel for cleaning the drill hole. If required, by turning the throttle *j* backward, live air can also be delivered through the bit, by way of the port *p*. Weight of drill, 50 lbs.

Murphy Drill is another example of valveless machine, made in both the D-handle and air-feed types, by C. T. Carnahan Manufacturing Co., Denver, Colo.

Ingersoll-Rand " Jackhamer " is made in two styles: " BCR-430," for dry holes; " BCRW-430," for wet holes. Referring to Fig. 158, *a* is the hammer, *b* the ratchet, *c* the rifle-bar, *d* the retaining bushing for the hexagonal drill shank. The forward end *e* of the hammer has straight longitudinal fluting, which engages with corresponding grooves in the bushing *d*, and thus rotates the bit on the back stroke. Fig. 159 shows the assemblage of parts of the rotation device, lettered as above. The valve is the same as described under the Ingersoll-Rand " Butterfly-Valve " drill (Chap. XX). An automatic oiler is shown at *h*. By the pulsations of the inlet air, oil is drawn into the valve chest, passing thence into the cylinder. Operation of the drill is eased by the heavy springs *f,f*, connecting the front and back cylinder heads. The bit is held firmly in position in the chuck by pressure of the double spring *g*, but is readily changed. The handle has rubber grips.

The Water Jackhamer " BCRW-430" (Fig. 160) is like the above, except that it has a central water tube for delivering water under pressure to the hollow bit, as in the Leyner-Ingersoll drill, already described. It can drill holes to a depth of 10 or 12 ft. The water tube is supplied with water through a swivel connection and a strainer. As the handle is offset, the tube is

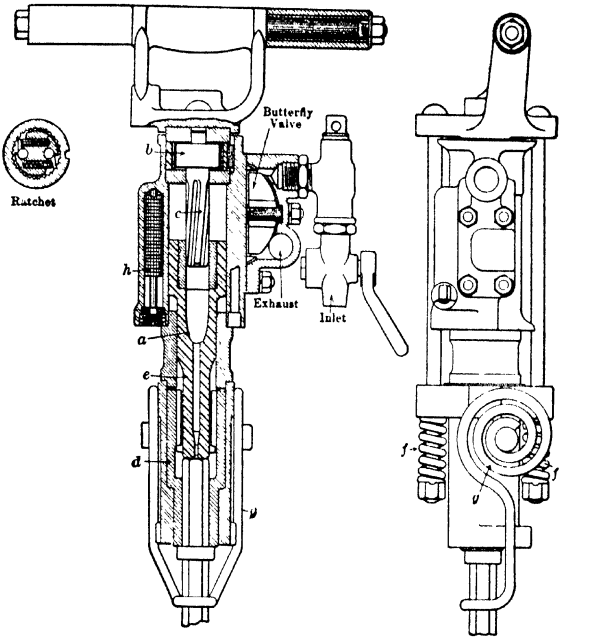


FIG. 158.—Ingersoll-Rand "Jackhamer" Hand Drill, "BCR-430," for Dry Holes.

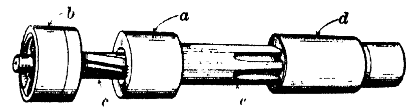


FIG. 159—Rotation Device of "Jackhamer" Drill.

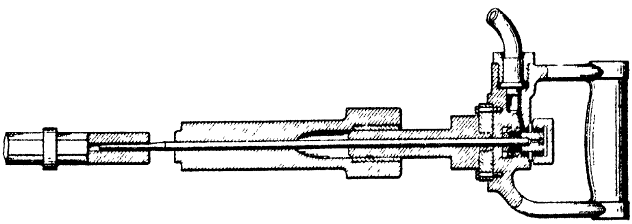


FIG. 160.—Ingersoll-Rand "Water Jackhamer," for Wet Holes.
(Cylinder and accessory parts not shown.)

readily removable. A standard " BCR " machine can be converted into a " BCRW " water drill by substituting a few parts. The water pressure should be at least 25 or 30 lbs., but must always be less than the air pressure. Water tanks of 6 or 18 gals. capacity are furnished by the makers. The bit shank of the hollow steel for the Jackhamer drills is shown

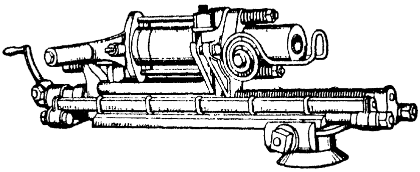


FIG. 161.—" Jackhamer " Drill on Cradle Mounting

in the second cut of Fig. 153. Both of the above machines weigh 41 lbs. Diameter of steel, $\frac{7}{8}$ in.; air hose, $\frac{3}{4}$ in.; water hose, $\frac{1}{2}$ in.

For flat-hole work, like drifting or breast-stoping, the Jackhamer may be mounted as shown by Fig. 161. The cradle,

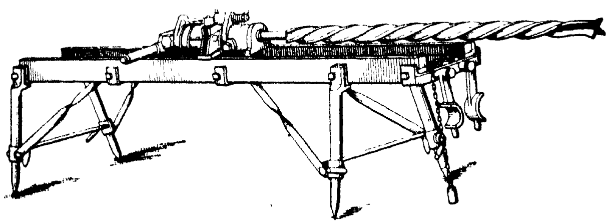


FIG. 162.—" Jackhamer " on Special Mounting for Thin Coal Seams.

somewhat resembling that of an ordinary rock-drill, is varied slightly in design according to whether it is to be used for the dry or wet machine; it is adapted to either column or tripod. A supplementary cradle, carrying the drill head, rides on the feed screw of the main cradle. Weight of mounting, 63 lbs.

For mining thin coal seams the " Jackhamer " drill is mounted as in Fig. 162. The legs of the mounting have adjust-

able extension ends, like those of a tripod. Cruciform steel, with a cross bit, and twisted as in Fig. 163, is used instead of ordinary octagon steel. When drilling, the machine slides forward on the wooden frame, the bit being supported and held in alignment by a pair of guide clamps pivoted on the front end of the mounting. As the auger-like steel rotates, it assists in removing the cuttings from the hole. This new mode of mounting is well adapted to drilling the breast holes so common in coal mining.

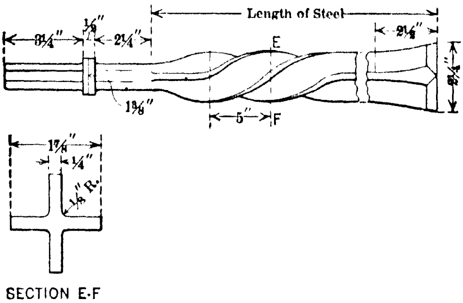
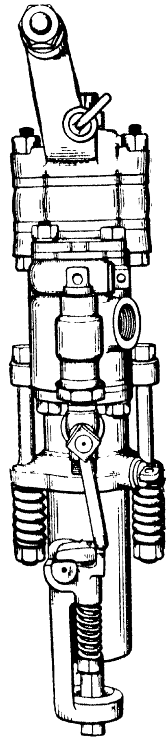


FIG. 163.—Twisted Cruciform Bit, for Drilling in Coal.

Ingersoll-Rand " Jackhamer Sinker " (Fig. 164) is a recent variation of the " Jackhamer," especially suited to shaft-sinking, or where deep holes (to 12 ft.) are required. It is built in two types, dry and wet, both fitted with an axial tube for cleaning the hole with either air or water; in other respects they are identical in construction. In the dry machine, the air delivered to the tube is controlled by a valve in the handle. The water device is the same as in the Leyner-Ingersoll drill, and the rotation like that of the standard " Jackhamer." The valve works on the principle of the " Butterfly," but is cylindrical, with end seats. As shown in the cut, the bit holder has a new form; to remove the bit, the yoke is swung to the right. Weight of drill, 70 lbs.; size of steel, 1 in.

Ingersoll-Rand " Bullmoose Jackhamer " is built especially for drilling deep down holes. Its design (Figs. 165 and 166)

is essentially different from the ordinary Jackhamer. A "Butterfly" valve is set at the back end of the drill, in line with the axis of the cylinder. The hammer reciprocates freely, the bit, which rests loosely in the chuck, being rotated independently,



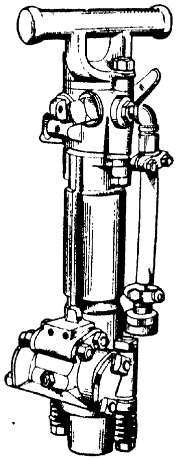


FIG. 164. "Jackhamer Sinker." FIG. 165.—"Bullmoose Jackhamer."

as shown in Fig. 166. A small spool-valve *a* is operated through auxiliary ports which are controlled by the movements of the hammer, and, by another set of ports, valve *a* throws the plunger *b*. A ratchet *c*, with four pawls, encircles the end of the bit chuck *d*, the ratchet in turn being inclosed by a steel ring *e*.

As the lug *f*, on ring *e*, engages with the plunger *b*, *e* is oscillated back and forth through a small arc, and by means of the ratchet thus rotates the bit.

This drill has automatic lubrication and a blowing device for removing sludge from the hole. One-inch hollow steel is generally used. Weight of drill, 105 lbs.

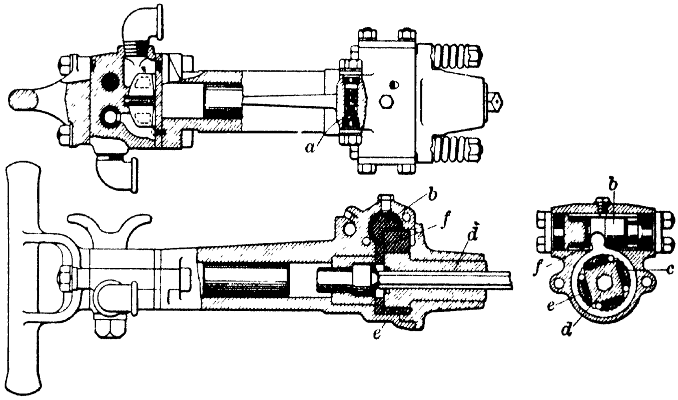


FIG. 166.—" Bullmoose Jackhamer."

Ingersoll-Rand " Imperial " Drills (types MV-1 and MV-2) are valveless (Fig. 167). They resemble in principle the Hardsocg drill, already described, in that the piston (or hammer) acts as its own valve. The hollow piston *a* has an enlarged part *b* near the rear end, against the shoulder of which the air pressure is constantly acting, and a series of 6 slot-shaped ports *c* near the forward end. In the cut the piston has completed its stroke; the air is being exhausted through the piston ports *c* to the exhaust port *d*. The return stroke is caused by the constant air pressure on the shoulder *b* of the piston.

The drill is rotated by a straight handle attached at *e*. The shank of the bit is held in a bushing *f*, which is made to receive either hexagonal or cruciform steel. Solid steel is used for these machines, which are designed for shallow " plug holes," for mining and quarry service. Weight of drill, 42 lbs.

For drilling $\frac{5}{8}$ to 1-in. holes to a depth of 6 ins., as for block-holing, pop-shots, dressing walls of shafts, cutting timber hitches, etc., the Ingersoll-Rand Co. makes a small "plug drill" (the "Invincible"), weighing $21\frac{1}{2}$ lbs. It is valveless, like the "Imperial." Downward pressure on the handle opens the throttle; the air is automatically shut off when the drill is raised from the hole. The steel is rotated by the handle or by a wrench. The exhaust is led through a hose to the mouth of the hole, for removing the cuttings.

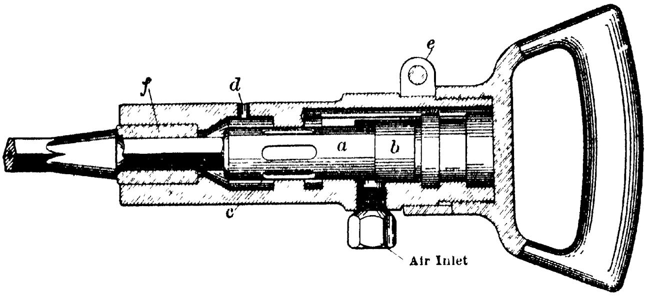


FIG. 167.—Ingersoll-Rand "Imperial" Hammer Drill (Types MV-1, MV-2).

Sullivan "Rotator" Hammer Drills are made in two forms, *i.e.*, with air-tube or water-tube.

The "Air-Tube Rotator" (Fig. 168) has a spool valve *a*, in a chest forming part of the cylinder casting. The rotation device is similar to that of the Sullivan "DR-6" drill (see description accompanying Fig. 154); ratchet *b* (Fig. 168) encircles the hammer *c*, which, by means of both straight and spiral grooving *h* and *i*, rotates the chuck bushing *d*. Air enters to the throttle at *e*, part of it going by the tube *f* to a passage in the rear cylinder head and thence through the axial air-tube *g* to the hollow bit. The bit is held in the chuck by the spring yoke *j*. Under the handle is an automatic lubricator *k*, like that used for the Sullivan "Liteweight" and "Hyspeed" drills. With each pulsation of air in the rear end of the cylinder, one of the small balls admits air to the oil chamber, *l*, and the other

ball discharges a little oil into the cylinder, in the form of spray. Weight of drill, 38 lbs.; steel, $\frac{7}{8}$ in.

The "Water-Tube Rotator" (Fig. 169) resembles the "Air-Tube" machine so closely that a description is unnecessary. It is intended especially for rather deep holes pointed

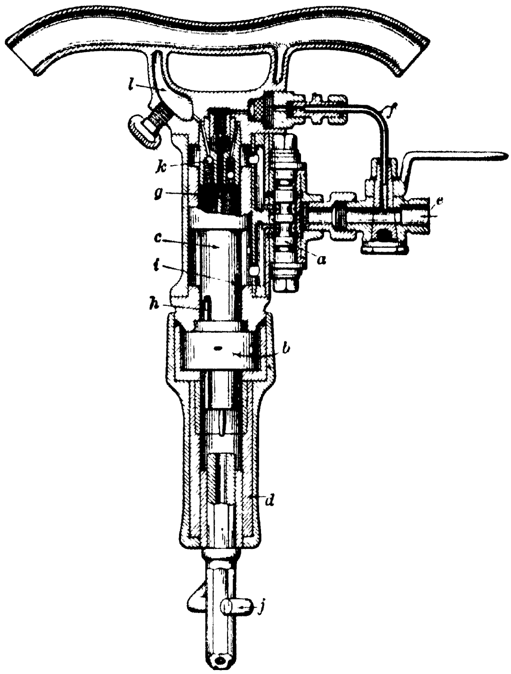


FIG. 168.—Sullivan "Air-Tube Rotator" (DP-33)

downward. Water, supplied by a tank under air pressure, like that in Fig. 136, Chap. XX, is admitted through the rear cylinder head to the axial tube. An inlet screen keeps out dirt and grit. A jet of live air mingles with the water, to increase the cleaning action. Weight of drill, 40 lbs.; steel, $\frac{7}{8}$ in.

Sullivan "Auger Rotator" (Fig. 170) is similar to the "DP-33" drill, but has a shorter, lighter and faster stroke. It

is designed for drilling in soft or broken rock (shales, coal, etc.). Solid, spiral steel is used, with a forked ("fish-tail") bit. It will drill 6–8-ft. holes in soft ore or coal, and up to 12 ft. in loose material. The piston is solid, no air or water tube being required, as the rotation of the spiral steel removes the cuttings from the hole. Weight of drill, 39 lbs.

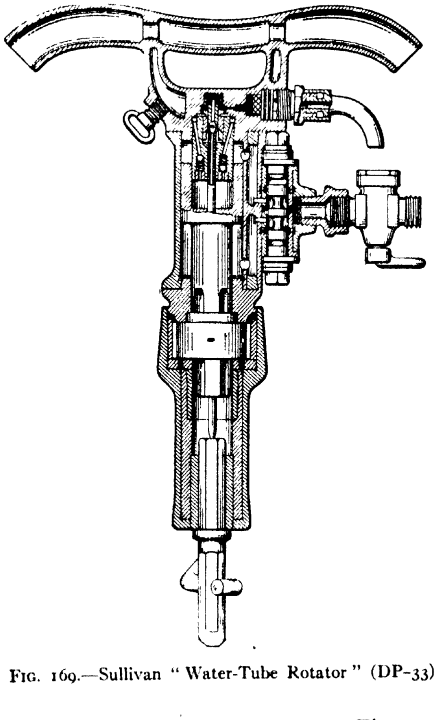


FIG. 169.—Sullivan "Water-Tube Rotator" (DP-33).

Mountings for Sullivan Hammer Drills. Fig. 171 shows the "Rotator," with handle removed, mounted on a cradle attached to a column arm. The special cradle has two clamps; one (*a*) fitting over the cylinder, the other (*b*) bearing on the socket of the drill handle. A coil spring (*c*) at the rear of the shell acts as a shock absorber. Weight of cradle, 60 lbs.

Another form of mounting, the " pneumatic feed " (Fig. 172), consists of a light column, on the arm-saddle (*d*) of which is clamped a trunnion (*e*), with a hinged cradle-clamp, supporting a long air-feed cylinder (*f*). On the forward end of the piston rod of (*f*) is a short arm (*g*) and a saddle to which the drill is

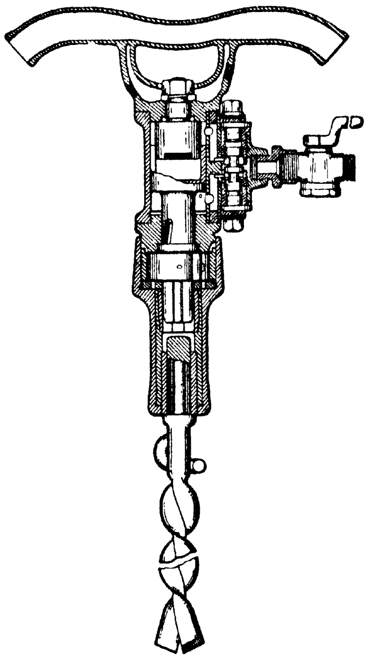


FIG. 170.—Sullivan " Auger Rotator," Class DR-33.

clamped. The action of both drill and feed is controlled by a single throttle.

" **Hummer** " **Drills** (Chicago Pneumatic Tool Co.). These are hand machines, with automatic rotation. Fig. 173 shows longitudinal sections of type " A-86," at right angles to each other.

Valve mechanism (Fig. 174). The hardened and ground steel ball B, weighing approximately 1 oz., reciprocates $\frac{3}{32}$ in. in a steel cage V, provided with end seats. S is the inlet port to the valve chamber, C and P are the cylinder and hammer, S_1 and S_2 the cylinder inlet ports, and E_1, E_2 the exhaust ports.

Compressed air enters V through a series of peripheral holes H, as shown by the arrows. The valve will thus be thrown

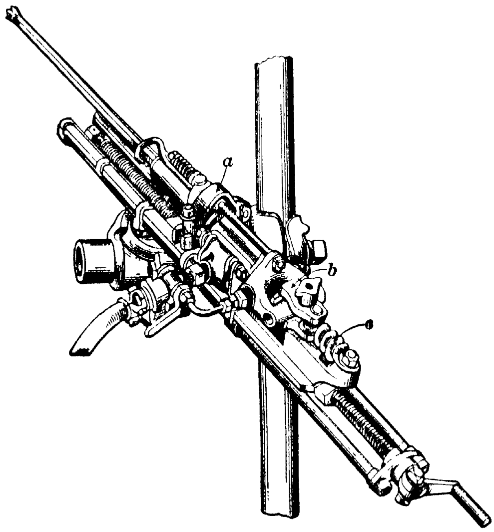


FIG. 171.—Cradle and Column Mounting for Sullivan "Rotator" Drill.

either forward or backward. Assuming that it takes its initial position as in the diagram, air flows through S_2 to the cylinder, and the hammer P makes its backward stroke. This movement of P uncovers exhaust port E_2 and covers E_1. When air is exhausted through E_2, an unbalanced condition is produced in the valve cage V, which causes valve B to move to its right-hand seat. Air from S then passes through S_1 into the cylinder, and the hammer makes its forward stroke, thus completing the cycle of operation.

Rotation of the bit is independent of the hammer. Referring to Fig. 173, a small, high-speed, rotary air motor *a* is set trans-

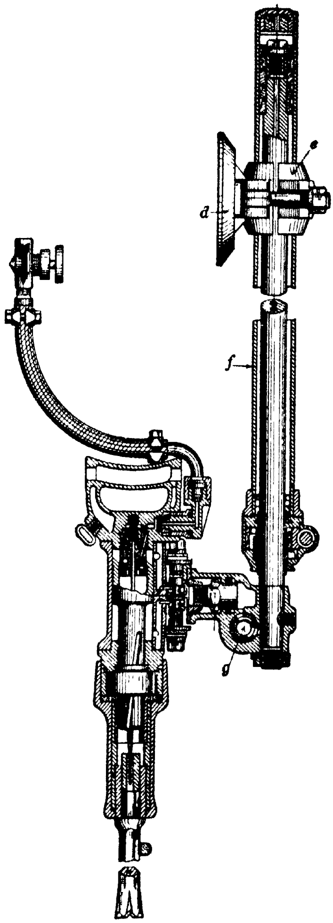


FIG. 172.—Pneumatic-feed Mounting for Sullivan "Rotators."

versely across the back cylinder head. The worm thread *b*, on the rotor shaft *c*, engages with a worm gear on the longitudinal

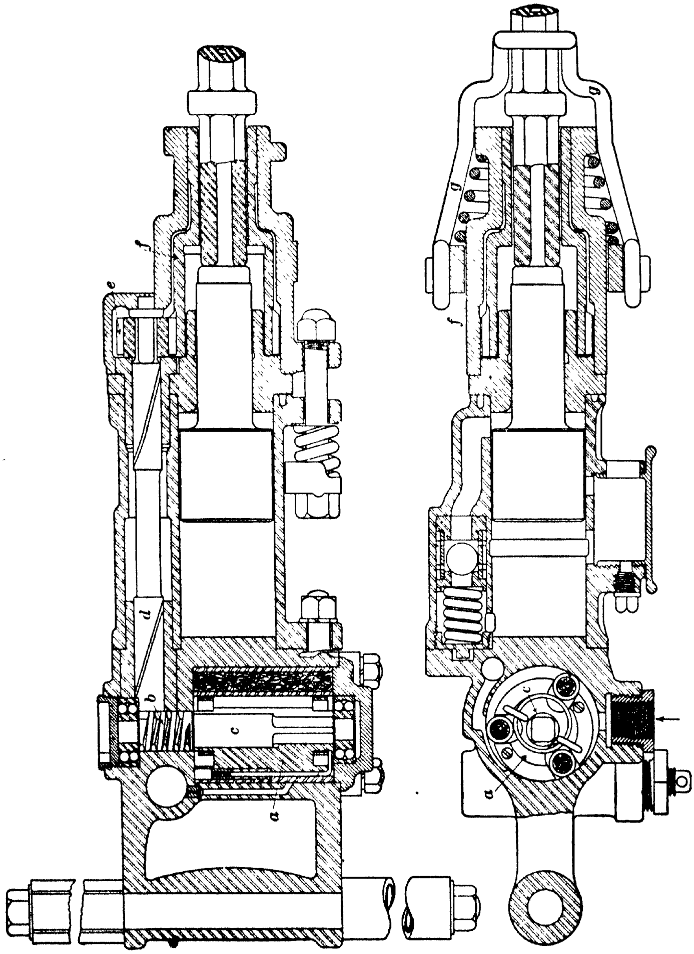


FIG. 173.—"Hummer" Drill Chicago Pneumatic Tool Co.

shaft *d*. At the other end of *d* is a pinion *e*, meshing with a gear mounted on and encircling the chuck bushing *f*, which holds the hexagonal shank of the bit. There is no ratchet. The air admitted to the drill goes first to the rotation motor; the exhaust from this passes through *S* (Fig. 174) to the valve *B*, for operating the hammer. The bit holder *g* (Fig. 173) is similar in general design to that of several of the machines previously described.

For drilling in coal, slate, or other soft rock, a twist or auger bit may be used (see Ingersoll-Rand " Jackhamer " and Sullivan " Auger Rotator ").

The Chicago Pneumatic Tool Co. makes several other types of hammer drill.

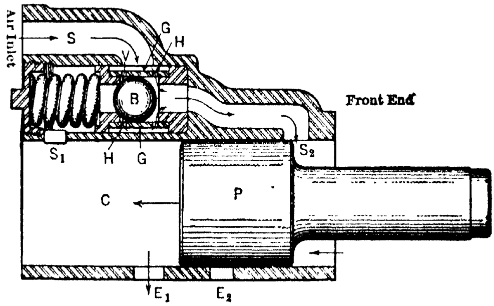


Fig. 174.—Valve Mechanism of " Hummer " Drill (Diagrammatic).

Waugh Hammer Drills (Denver Rock Drill Manufacturing Co.) are made of several types. (Waugh " Stopers," with telescopic air-feed standards, are described later.) The " Clipper " drill, unmounted, weighs 47 lbs.; the " Dreadnaught," 83 lbs. Their general design is essentially the same. As hand drills, they serve for shaft-sinking, blockholing, quarrying, etc.; mounted on a guide shell (on column or tripod), they are applicable to horizontal breast work and drifting.

The " Clipper " drill, Model 50 (Fig. 175), is valveless. Air enters at *a*. The hollow hammer *b* has four ports, as shown. In making the forward stroke, air acts on the entire area of the hammer; for the back stroke, it acts on the shoulder *c*. Rotation

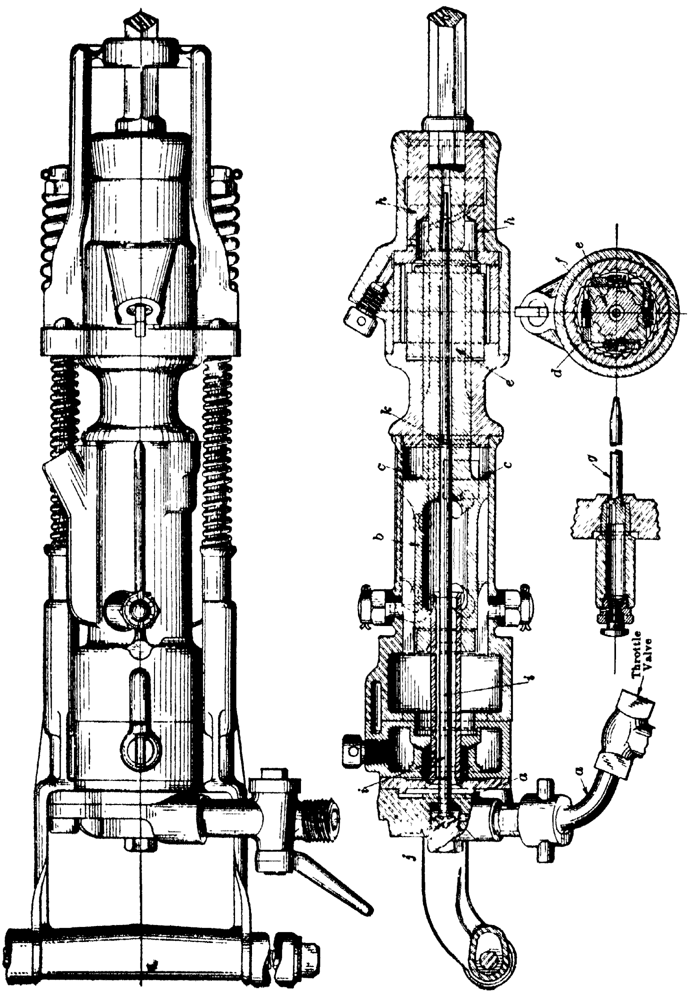


FIG. 175.—Waugh "Clipper" Drill, Model 50.

is effected by the ratchet *d* (in the front cylinder head), which encircles the rifled extension *e* of the hammer. As the hammer is prevented from rotating by two longitudinal splines in the cylinder, each stroke of the hammer turns the rifle-nut *f*, which contains the pawls and their springs. In the forward edge of the ratchet ring *d* is a set of small lugs, which engage with lugs on the chuck bushing *h*, holding the hexagonal bit shank.

The water tube *i* receives water at *j*, the supply being controlled by the needle valve *g* (see detail cut). As the tube is reduced in size at the point *k*, making a loose fit thence to the end of the hammer, air mixes with the cleaning water. To increase the quantity of water, when required, the drill is raised slightly, allowing the hammer to run forward and thus uncover more of the smaller part of the tube *i*. Model 55 of the Waugh drill has no water attachment, a blow-valve being used instead, to deliver a large quantity of air at intervals, when drilling deep holes.

The "Dreadnaught" hand-hammer drill (Model 60) is similar in general design to the "Clipper," but is larger and heavier. Another type of "Dreadnaught," having a valve and an air-feed standard, is described on p. 340.

McKiernan-Terry "F-I" Hammer Drill (Fig. 176) has automatic rotation, uses hollow, $\frac{7}{8}$-in. steel, and will drill to a depth of 8 or 10 ft. Weight, 38 lbs. The spool valve *b* is small and light, with a throw of only $\frac{1}{8}$ in. The throttle *a* has three positions: the first shuts off air; the second opens the port *c*, to admit air to the hollow bit, for cleaning the hole; the third starts the drill. When blowing out the drill hole, the bit is slightly raised from the bottom. From port *c* the air for cleaning goes to the forward end of the cylinder through passage *d*, alongside of the hammer. Rotation is similar to that of the Leyner-Ingersoll drill (Fig. 152); *e* is the rifle-bar, *f* the rifle-nut, in the hammer *g*; the front end of *g* has longitudinal fluting *h*, which engages with corresponding grooves in the chuck bushing or socket *i*. This bushing receives the hexagonal shank of the bit. To remove the bit, the yoke *j*, normally held in position by the springs *k*, is swung to the right.

McKiernan-Terry " A-9 " Hammer Drill (Fig. 177), weighing 90 lbs., is designed for drilling deeper and larger holes than the preceding. The spool-valve is like that of many of the standard reciprocating drills. Aside from this, though there are

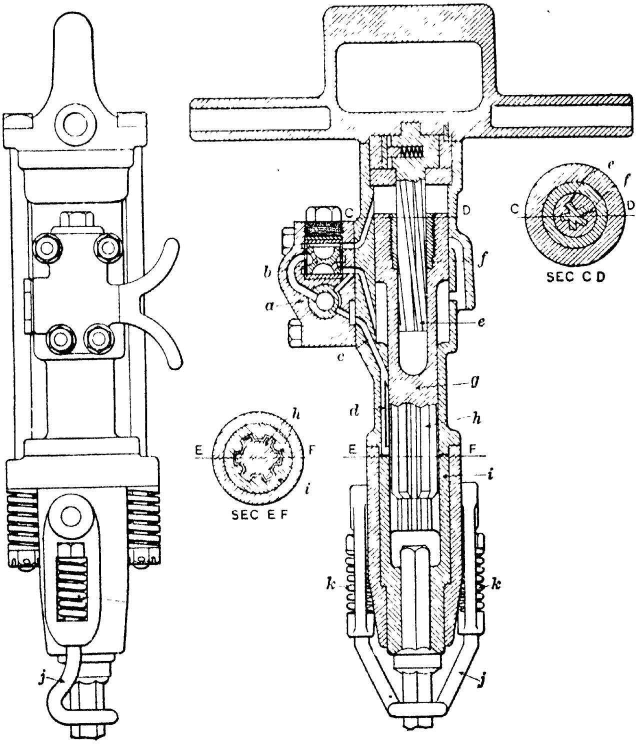


FIG. 176.—McKiernan-Terry " F 1 " Hammer Drill.

differences in the details of construction, the main features are the same as in the " F-1 " drill. Size of steel used, 1 in. The steel is not specially shanked, the rotating chuck bushing receiving the full hexagonal section.

Another drill by the same makers is the "Busy Bee " (" B-1 ")

weighing 50 lbs. It is designed for shallower holes, and for dressing or trimming, blockholing and similar work.

Wood Hammer Drill has a spool-valve, of essentially the same design as the Wood reciprocating drill (Chap. XX). In its general features, including the mode of rotating the bit, it is quite similar to the McKiernan-Terry "A–9" drill, described above. Weight, 44 lbs.; diameter of cylinder, 2 ins.

CLASS C. STOPERS, OR HAMMER DRILLS WITH AIR-FEED STANDARDS

General Description. These machines are made by most of the rock-drill manufacturers. They are intended chiefly for drilling holes directed above the horizontal, as in overhand stoping, though they may also be mounted on a column, for drifting and other breast drilling.

Nearly all makes are built on the same general lines. The design of the drill itself is usually the same as that of the hand-hammer drills of the same maker. Attached to the back head of the drill is a long, telescopic extension, on the end of which the machine stands when drilling overhead holes. This standard comprises a slender cylinder and piston, which when supplied with compressed air automatically keep the drill fed up to its work. Air for both drill and feed cylinder is admitted by a single throttle valve.

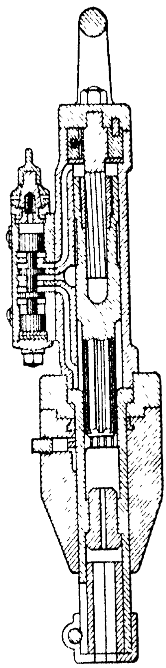


FIG. 177. — McKiernan-Terry "A–9" Hammer Drill.

Most of these drills have no automatic rotation; to keep the hole round, the entire machine is rotated on its axis by an arm or handle. They are all one-man machines. A few makes use a water attachment, for directing a spray of water at the mouth of the hole, to moisten the dust.* Others use hollow

* In this connection, see the "dust allayer" of the *Climax* drill, p. 280.

steel, with a water tube passing through the axis of the drill, like many of the drills in Classes A and B.

Owing to the similarity in design of the air-feed of the different makes, and to avoid repetition, the reader is referred to the following cuts for details of construction. The descriptions of typical drills given below are confined chiefly to the valve motion and the operation of the drill itself.

Waugh Stoper is made in several sizes and weights. The lightest model ($2\frac{1}{8}$-in. cylinder), using 1-in. steel and striking a short, rapid blow, is suitable for the softer rocks and ore. The heaviest machine ($2\frac{3}{4}$-in.) has longer stroke, uses $1\frac{1}{8}$ or $1\frac{1}{4}$-in. steel, and strikes a heavier blow, as needed for hard ground.

A recent model of the Waugh stoper is the No. 14A ($2\frac{1}{2}$-in.). Its operation is shown by Figs. 178, 179 and 180. The valve *A* is a hollow cylinder, with external annular recesses and a rear collar *r*. On the back stroke, the neck *w* of the hammer enters the bore of the hollow valve. Compressed air is admitted to the drill by the path shown by the arrows, passing through the channels *a*, *b*, *c* and *d*, to the valve, which then takes the rearward position, as in Fig. 178. This position is caused by a differential pressure on the valve, due to the small difference in the diameters and areas of the portions *y* and *z*. The live air from the recess or chamber *c* is then free to flow into the rear end of the cylinder *e* (Fig. 179), since the diameter of the hammer neck *w* is about $\frac{1}{32}$ in. less than the diameter *x* of the valve chest. As the hammer is driven forward, any air in the forward end of the cylinder remaining from the preceding stroke is exhausted through ports and passages *g*, *h*, *i*, *j* (Fig. 180), annular groove *k* of the valve (Fig. 178), and exhaust ports *l* and *m* (Fig. 179), to the atmosphere.

When the hammer in its forward stroke uncovers the opening *n* (the "trip-hole") (Fig. 180), air flows through passages *n*, *o*, and *p*, into chamber *q*, where it acts on the large end *r* of the valve and shifts it into its forward position. Air then passes from *b*, through *d* and recess *k* (Fig. 178), and thence through *j*, *i*, *h* and *g* (Fig. 180) to the front end *f* of the cylinder, thus driving the hammer back. During this stroke, the air in the

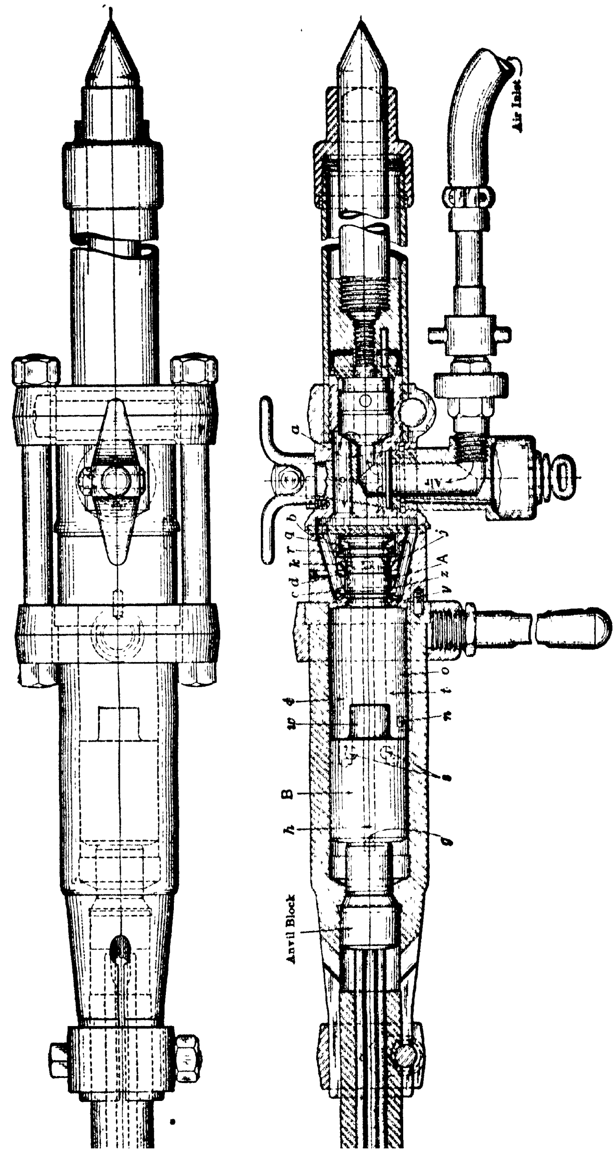


FIG. 178.—Waugh Stoper, Model 14 A (Denver Rock Drill Manufacturing Co.)

rear end of the cylinder is exhausted through the hollow valve into chamber *q* (Fig. 179), and thence out of the exhaust ports *l* and *m*. As the exhaust is not yet down to atmospheric pressure, it holds the valve in its forward position, but, when the hammer neck *w* enters the valve, the exhaust pressure in *q* drops to atmospheric pressure, and no longer holds the valve. There-

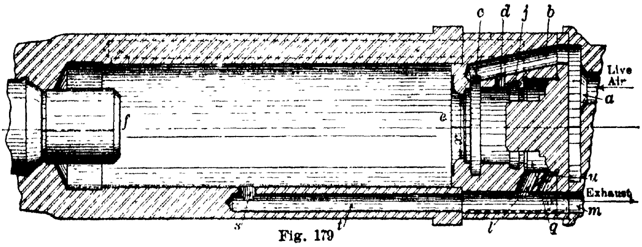


Fig. 179

Note These Sections are Diagrammatic; Hammer not shown. There are actually 2 Ports *g* and *h*; 6 Ports *d* and *b*. 4 Ports *m* and 4 Ports *l*.

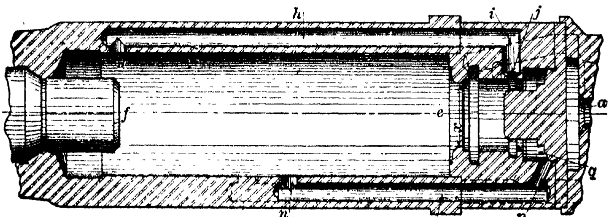


Fig. 180

Figs. 179 and 180.—Waugh Stoper, Model 14A.. Diagrammatic Sections of Cylinder, Valve and Ports.

upon, the differential pressure on areas *y* and *z* (Fig. 178) becomes effective, and the valve shifts to its rearward position. Air then enters the rear end of the cylinder at *e*, and the hammer is again driven forward, thus completing the cycle. Near the end of both forward and back strokes air exhausts expansively through ports *s*, *t* and *m* (Fig. 179). The small passage *u* is a vent for the chamber *q*.

Ingersoll-Rand Stope Drills are of two types, the chief difference being in the design of the cylinder. The air feed, made in three forms, is the same for both, as noted below.

The "Butterfly" stoper (Fig. 181) is of solid bolted construction, except the front head, which is spring retained. An "anvil block" *a* is interposed between the hammer *b* and the drill bit *c*. Air is admitted from the inlet *d* by the throttle *e* to the "Butterfly" valve *f*, the ports being shown by the dotted areas opposite each wing of the valve. One wing controls admission to both ends of the cylinder, the other controlling the exhaust. Details of the operation of this valve are given in Figs. 141–143 (Chap. XX), with accompanying descriptions. The exhaust opening *g* directs the exhaust backward, to minimize its tendency to scatter the dust at the mouth of the drill hole. Throttle *e* is so designed that, when in position to admit air to

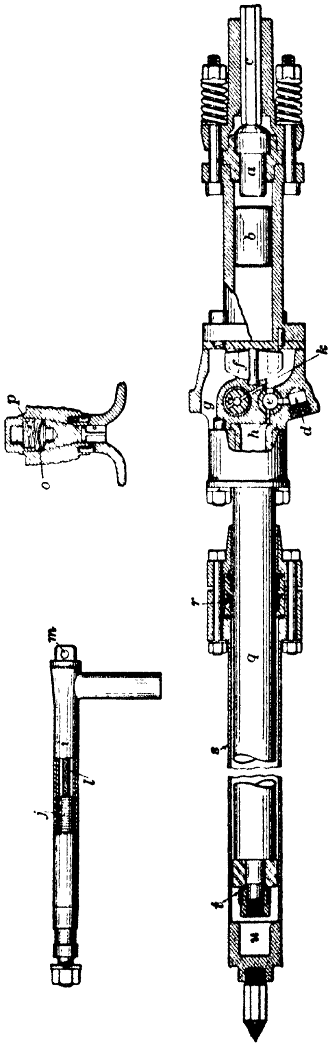


FIG. 181.—Ingersoll-Rand "Butterfly" Stoper.

the drill (as in Fig. 181), the small port *h* is also opened, to allow air to pass to the feed cylinder.

The rotating handle, shown in cross-section and detail drawing at *i* (Fig. 181), and in longitudinal section by Fig. 182, contains an automatic oiling device, operating as follows: In the bore of the handle is a porous plug *j*, which regulates the flow of oil and strains out dirt and grit. The oil chamber *l*, filled once a shift by removing the screw plug *m*, is in communication by the small port *k* with the live air side of the valve chest. When the drill is at work, the air pulsations in the chest draw the oil through *j* and the small passages *n* and *k* into the chest, whence it passes with the air into the drill cylinder. In the valve chest, close to the throttle, is an air strainer *o*

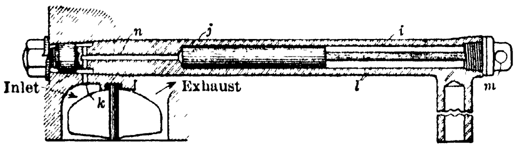


Fig. 182.—Oiling Device, "Butterfly" Stoper

(see detail cut in Fig. 181), consisting of a perforated cup-shaped disk *o*, held in position by the spring *p*.

Three forms of air feed are made: (1) the cylinder feeds off the piston (Fig. 181), which allows the drill to be mounted on a column or tripod, for making breast holes; (2) the piston feeds out of the cylinder; (3) an extension point is added, for drilling uppers in workings with a high roof. Referring to Fig. 181, the hollow piston *q* is a steel tube, with a flange for bolting to the drill body. The stuffing box *r* of the cylinder *s* contains two cup-leathers. When the piston and cylinder close up, they are automatically held together by the friction spring *t* entering the recess *u*. Fig. 183 shows the air feed of the second type.

The "Butterfly" stoper uses 1, $1\frac{1}{8}$ or $1\frac{1}{4}$-in. cruciform steel, (generally solid), with square, hexagonal or cruciform shank. Weight of drill, 74 lbs.

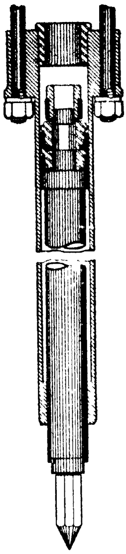


FIG. 183.—Air Feed for "Butterfly" Stoper, Type "BC 21"

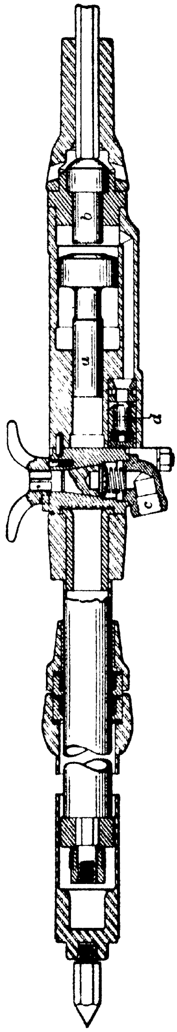


FIG. 184.—Ingersoll-Rand "CC Stopehamer" (Dry Type).

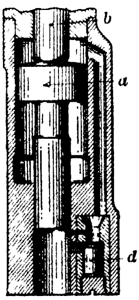


FIG. 185.—Cylinder and Valve of "CC Stopehamer."

The Ingersoll-Rand " CC Stopehamer " (Fig. 184) is similar to the " Butterfly " stoper, except in the design of cylinder and valve. The cylinder is a drop forging; *a* is the hammer, *b* the " anvil block," *c* the inlet elbow. The valve *d* has a very short throw and is designed to give a high degree of expansion in the use of the air; it has end seats. Fig. 185 shows the cylinder, valve and ports in more detail.

A wet type of " Stopehamer " (CCW) is also made, the water device being the same as in the " Butterfly " stoper. Fig. 186 shows a spray dust-allayer, which may be attached to

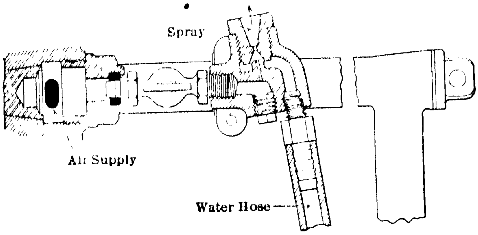


FIG. 186.—Dust Allayer for " Stopehamer."

the " CC " drill by replacing the inlet elbow by a special connection. Weight of the " Stopehamers " is 81 lbs.

" Chicago Stoper " (Chicago Pneumatic Tool Co.), with air-feed standard, has the same valve motion as the " Gatling " drill of the same makers (Chap. XX, Fig. 144). As it resembles in its general features the drills of this class described on preceding pages, details are omitted. It ranks with the best of the stoping drills. Solid, cruciform steel is used, and a spray dust-allayer is provided, which is readily attached when desired. Weight of drill, 70 lbs.; diameter of cylinder, $2\frac{1}{8}$ ins.

Sullivan Hammer Drills with air-feed. In these the feed cylinder is not an integral part of the drill. For details and illustrations, see Sullivan drills under Class B.

Cochise Air-Feed Drill (Cochise Machine Co., Los Angeles, Cal.) is similar in general design to the other typical stopers with air-feed standards, as described on preceding pages. It

has a spool valve. Weight of drill, 73 lbs.; diameter of cylinder, $2\frac{1}{8}$ ins.

OPERATION OF HAMMER DRILLS

Air Consumption of Hammer Drills is approximately the same as for reciprocating drills of the same cylinder diameter (see Tables XXXI and XXXII, Chap. XX). But, in comparing the results in terms of work done, it must be remembered that the smaller sizes of Class A drills, and all of Classes B and C, are one-man machines, and that for some kinds of work, especially overhead stoping, the hammer drills make a higher footage of hole.

Depth of Hole. When hammer drills were first introduced, it was found that the speed of drilling materially decreased at depths greater than 3 or 4 ft., even with the use of a water jet alongside of the bit. This was due in part to the inertia of long and heavy bits, but probably more to the failure of the early drills to " mud " well. In recent years the designs have been so improved that, even for " down " holes, they can now be used for depths of at least 10 ft., though average depths are usually from 4–6 ft. This is largely the result of using hollow steel and a strong jet of water mixed with compressed air. The force of the expanding air assists in keeping the hole clear of pasty sludge, thus allowing the hammer to strike a more effective blow. As a rule, the fastest work is done when drilling " uppers " (holes directed at a steep upward angle) in dry rock or ore. The dust and cuttings then run out by gravity; that is, the holes are self-cleaning.

Records of Work. The prefatory remarks, made under this heading in Chap. XX, respecting reciprocating drills, apply also here. But, since hammer drills are usually operated without mounting, no allowance of time for setting up is necessary; and, as there are no chuck bolts to manipulate, bits can be changed in $1\frac{1}{2}$–2 minutes. Table XXXVIII gives figures based on a number of recorded runs. Type of drill (column 3) refers to the classification used in this chapter.

TABLE XXXVIII

Kind of Work.	Rock or Ore.	Type of Drill.	Depth of Hole, Ft.	Inches of Hole per Minute.	
				Total Time	Net Time.
Stoping	Hard phonolite breccia	Class C	2 2	2 04	2 54
Same as above, but unfavorable conditions			2 6	1 08	
Drifting.	Amygdaloid copper rock.	Class A	6 0	0 72	
Drifting	Tough schist.	Class A		2 64	
Stoping	Hard trachyte.	Class C	4 0	2 35	2 90
Stoping	Hard trachyte.	Class C	5 0	1 20	
Stoping	Hard porphyry	Class C	5 5	2 09	
Stoping	Amygdaloid copper rock.	Class C	6 0	0 93	
Drifting.	Amygdaloid copper rock.	Class A	6 0	2 15	3 22
Stoping	Quartzose ore	Class C	3 0	2 15	3 58
Drifting	Granite.	Class A	3 0	3 00	
Drifting.	Quartzose ore.	Class A	3 3	1 58	
Stoping	Medium andesite.	Class C	4 3	2 85	3 52
Stoping	Pyritic ore, medium hard	Class C	5 3	2 00	
Stoping	Pyritic ore, medium hard	Class C	5 0	1 10	
Tunnelling (test run)	Average granite.	Class A	7 0	3 56	5 16
Sinking	Tough schist	Class B	6 5	1 43	
Sinking	Tough schist	Class A	7 7	1 00	
Drifting.	Hard limestone.	Class A	3 2	1 63	1 86
Raising	Quartzose ore	Class C	4 0	2 40	5 10

Field of Work. For driving tunnels, drifts and crosscuts, or for underhand stoping in wide veins, and wherever deep holes of large diameter can be advantageously adopted, reciprocating drills are still in general use, though the same field of work is occupied by the hammer drills of Class A. In connection with these operations, hammer drills are useful as auxiliaries, for blockholing and for "squaring up" after the main rounds have been fired; that is, dressing the walls and taking up the "bottom" when the deep holes fail to break clean.

Most of the hammer drills, particularly those of Classes B and C, are less well adapted to making holes that approach the horizontal, as in tunnelling, drifting, crosscutting and breast stoping. Class B machines are best for down holes, as in quarrying and trenching in rock. They are also used success-

fully for shaft sinking in rather soft and laminated or thin-bedded rock (like shales), or where there are many slips and short fissures. In such rocks, a relatively large number of shallow holes give the best results, and to drill them with reciprocating machines involves extra loss of time for shifting and setting up.

Class C drills are especially designed for, and do their best work in, drilling uppers, as in general overhand stoping. They are useful, also, in mining thin veins with narrow paystreaks. For breast holes, they are sometimes mounted on columns.

In drilling dry holes, hammer drills raise much dust. With the hand machine, the operator must stand close to the mouth of the hole, and, when hollow steel is used with an air jet, the dust is blown back in the operator's face. This is especially troublesome in drilling uppers with Class C machines. The harmful effects of rock dust containing siliceous matter, also the spray dust-allayers and other water devices of hammer drills, have been referred to on preceding pages.

Solid bits have a limited application. They can be used for shallow holes in the softer rocks, by watering the hole and spooning out the sludge; or in dry rock, provided the holes are at a sufficient upward angle to permit the cuttings to run out by gravity. But they are best used for cutting hitches for mine timbers, blockholing, and quarry work.

Makers of Hammer Drills. The following alphabetical list includes all of the principal American makers:

C. T. Carnahan Mfg. Co., Denver, Colo.
Chicago Pneumatic Tool Co., Chicago, Ill.
Cleveland Pneumatic Tool Co., Cleveland, O.
Cochise Machine Co., Los Angeles, Cal.
Denver Rock Drill and Mach. Co., Denver, Colo.
Flottman & Co., Cardiff, Wales
Hardsocg Wonder Drill Co., Ottumwa, Iowa
Iler Rock Drill Mfg. Co., Denver, Colo.
Ingersoll-Rand Co., New York
McKiernan-Terry Drill Co., New York
Shaw Pneumatic Tool Co., Denver, Colo.
Sullivan Machinery Co., Chicago, Ill.
R. Stephens & Son, Carn Brea, Cornwall, England
Whitcomb Hammer Drill Co., Rochelle, Ill.

CHAPTER XXII

COAL-CUTTING MACHINERY

COAL-CUTTING machines have largely replaced hand labor in "under-cutting" the coal, preparatory to breaking it by blasting or wedging. Objects: (1) To economize in the cost of mining; (2) to decrease the proportion of "fines" produced; (3) to increase the rate of production from a given extent of mine workings. Coal cutters are used chiefly in bituminous collieries, and, unless wages are very low, they can mine more cheaply than by manual labor. In recent years they have also been successfully employed, to a limited extent, for mining anthracite in veins of rather flat pitch. They groove or undercut the face or breast of coal, close to the floor and to a depth of several feet. The mass so undercut is subsequently broken down by comparatively light blasts.

Coal cutters comprise four classes: (1) Endless-chain; (2) Rotary-bar; (3) Disk; (4) Reciprocating or pick machines. Machines of the first three classes are driven by electricity or compressed air, electricity being now in most general use. Pick machines are operated by compressed air, no satisfactory electric-driven pick having yet been brought out.*

Endless-Chain Cutters are built by several makers in the United States, among whom are the: Goodman Manufacturing Co., Jeffrey Manufacturing Co., Morgan-Gardner Electric Co., and Sullivan Machinery Co. Most of the machines of this type are electric-driven, though the Jeffrey and Sullivan chain cutters are furnished with compressed-air drive for use in gaseous mines, or where local conditions make it convenient.

* The "Pneumelectric Coal Puncher" (see p. 372) is not a true compressed-air pick machine, in the sense in which the term is here used.

The standard chain machines now in use are self-propelling along the face of coal to be undercut. Differences in design are chiefly matters of detail, rather than of principle. For shifting and feeding the machines, there are small wire-rope or chain drums, mounted on the rear end and operated by gearing from the air engine or electric motor. The rope or chain is made fast to a timber, or held by a " jack " set between

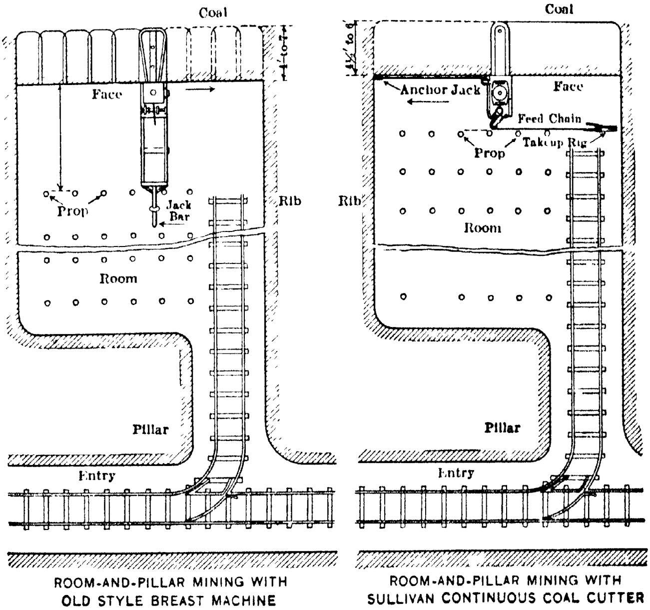


FIG. 187.

roof and floor, at any desired distance from the machine. By throwing the drum into gear, the machine is thus pulled (fed) along the face, as the undercut advances.

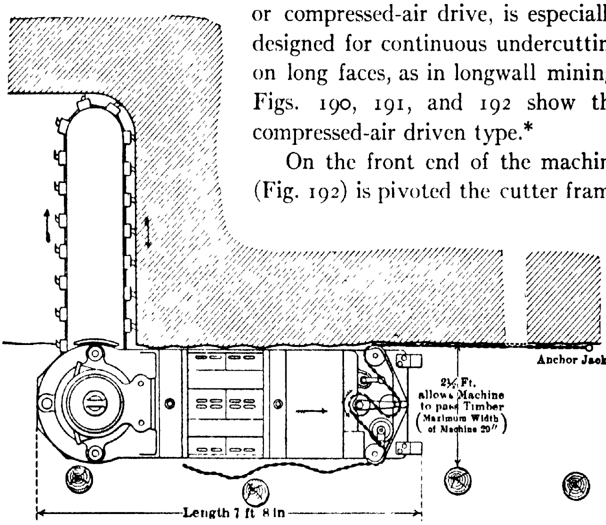
Some machines have but one drum; others (as the Jeffrey " Longwall ") have a special " sumping " drum, by which the machine is brought into position at the point where cutting is to be started, and is then fed forward or " sumped " into the face a distance equal to the depth of the slice to be taken.

Figs. 187, 188 and 189 show different modes of manipulating a chain machine in room work.

The makers of coal cutters furnish special trucks for moving the machines from place to place underground. Some of the trucks are propelled by a sprocket chain, driven from the cutter engine (see Fig. 193).

The Jeffrey "Longwall" chain machine, for either electric or compressed-air drive, is especially designed for continuous undercutting on long faces, as in longwall mining. Figs. 190, 191, and 192 show the compressed-air driven type.*

On the front end of the machine (Fig. 192) is pivoted the cutter frame



POSITION OF SULLIVAN LONGWALL IRONCLAD IN CUTTING ALONG THE FACE

FIG. 188.

a, carrying the chain. The chain runs on two sprockets, one at each end of the cutter frame (Fig. 190). At the inner end is the driving sprocket, operated through worm gearing by the compressed-air turbine *h* (Fig. 192). By means of the jaw clutch *f*, the cutter chain can be started or stopped, as required. The turbine (Fig. 191) consists of two rotors, with helical blades or teeth cast on their surfaces, and is designed to work with air pressures of 40–80 lbs.

* At the present time (1918), due to war conditions, the Jeffrey Co. is not building air-driven machines.

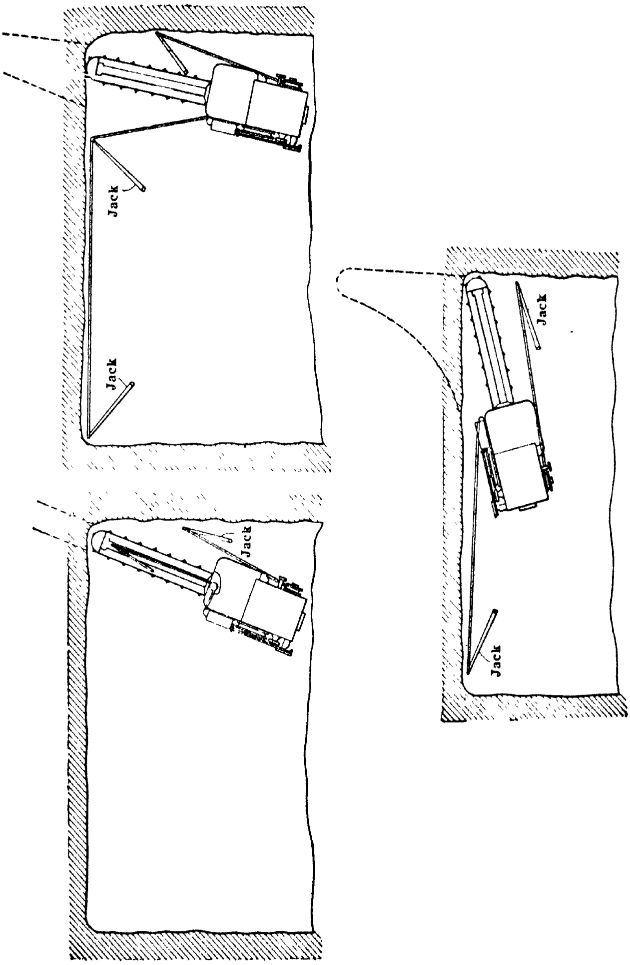


FIG. 189.—Jeffrey Coal Cutter. Three methods of anchoring and handling the machine in starting a cut.

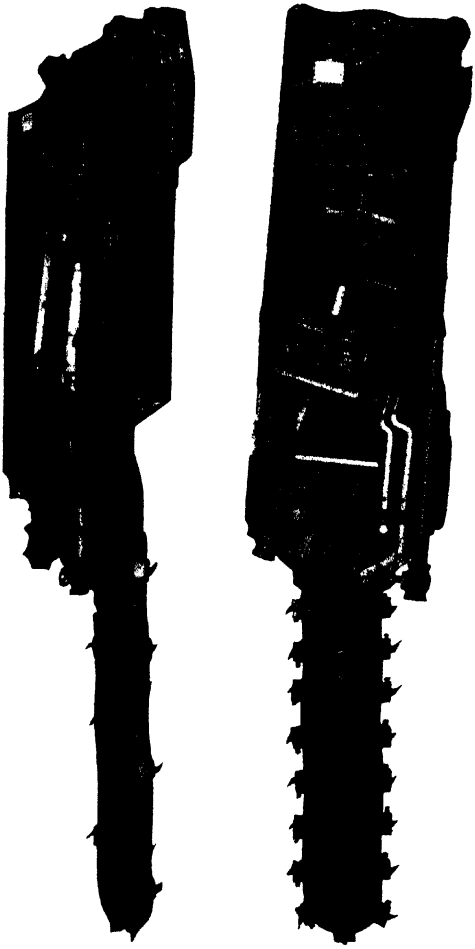


FIG. 190.—Longwall Air Turbine Coal Cutter, Type 24-A, Jeffrey Mfg. Co.

Another worm *i* (Fig. 192), on the opposite end of the turbine shaft *g*, operates the feed and sumping drums *j* and *k*, as follows. The feed worm-wheel *l* has a crank pin *m* and connecting rod

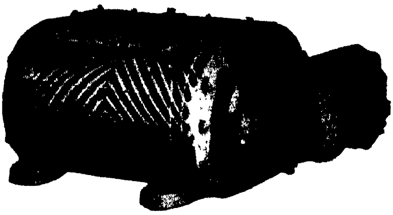


FIG. 191.—Compressed-Air Turbine for Longwall Coal Cutter, Jeffrey Mfg. Co.

n, which is pinned to the ratchet drive-lever *o*, carrying a pawl *p* engaging with the ratchet *q*. On the lower end of the vertical spindle *r* of this ratchet is a pinion *s*, engaging with a gear *t*,

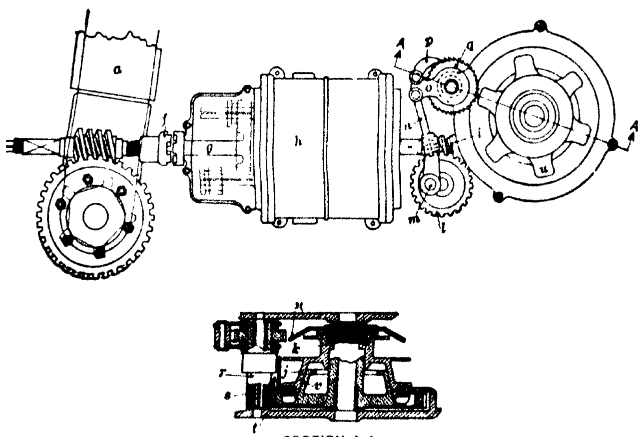


FIG. 192.—Diagram of Operating Mechanism, Jeffrey Coal Cutter, Type 24-A.

which is attached to the cone clutch *v*. The handwheel *u* operates the drums by forcing them down into contact with the cone clutch.

This machine is made in two sizes. The 24-A size is 8 ft. 2 ins. long, 31 ins. wide, 19 ins. high; weight, 6,000 lbs.; lengths of cutter arm, 4, 5, and 6 ft. The 36-A size, especially designed

FIG. 193.—Jeffrey "Shortwall" Coal Cutter, Type 35-B.

for thin, pitching seams, is 8 ft. 8 ins. long, 31 ins. wide, 15 ins. high; weight, 4,000 lbs.; lengths of cutter arm, 3, 4 and 5 ft.

The Jeffrey "Shortwall" Coal Cutter, type 35-B (Fig. 193), is built for both air and electric drive, and is designed especially

for room work. Fig. 189 shows the different modes of starting a cut. The cutter head is rigidly attached to the frame, the entire machine being swung in following its work. The chain is driven through worm gearing by a compressed-air turbine, similar in general to that used for the longwall cutter. Fig. 193 shows two views of this machine, as mounted on a self-propelling truck for delivery at the point where the work is to be done.

Fig. 194 shows an older type of Jeffrey chain machine, for room or breast work, and still used in a number of mines in the United States. It has a bed frame of two parallel steel channels, with cross braces, within which is a T-shaped sliding frame, carrying on the rear end a small duplex air engine. The sliding frame carries an endless sprocket-chain, driven by gearing from the engine. Sockets in alternate links of the chain hold the cutting teeth or bits. In the forward end of the sliding frame are two idler sprockets carrying the chain. On each end of the main frame are screw-jacks, for bracing the machine in position. The bits are so shaped and staggered as to "cover" the chain and cutter head, and make a groove in the coal about 4 ins. high, or sufficient to permit the cutter head to enter the undercut freely. The sliding frame is fed forward by a pair of pinions engaging with feed racks on the stationary frame, and driven by a worm gear from the engine.

Depth of undercut, 4-7 ft., width, 39-44 ins. By shifting the machine sidewise, successive cuts are made along the face. From 100-150 sq. yds. can be undercut in 10 hours. Power required, 8-14 H.P.

The Sullivan "Ironclad" coal cutters are built for both electric and compressed-air drive. Fig. 195 shows the longwall machine, class CH-8, operated by an air turbine. The Sullivan Machinery Co. also builds an air-driven chain machine for room-and-pillar work, class CE-7 (Fig. 195*a*). Figs. 187 and 188 show diagrammatically these two types, operating respectively in room-and-pillar and longwall mining.

In its general lines, the air turbine is similar to that used for the Jeffrey longwall machine, already described; it consists

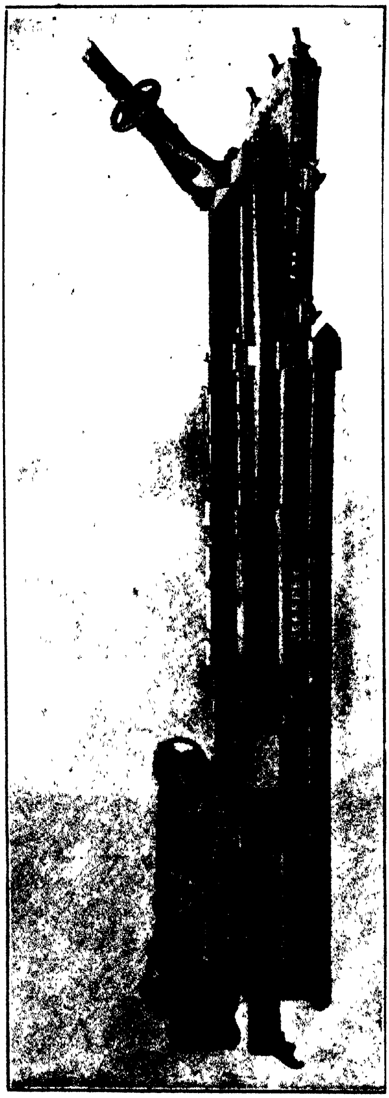


Fig. 194.—Jeffrey Chain Machine, Type 16-D.

essentially of a pair of wide helical gears, ground to each other and making a close fit in the casing.

Besides the machines already described, several other types of chain cutters are made in the United States, but, as they are

FIG. 195.—Sullivan "Ironclad" Longwall Coal Cutter. Class CH-8.

electric-driven, any discussion of them would be out of place in this book.

Rotary-Bar Cutters. A design formerly used to a limited extent in the United States for room work, but no longer made

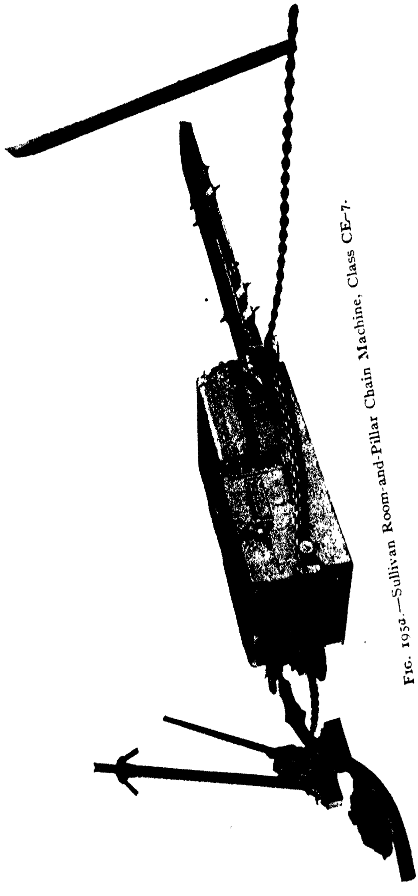


FIG. 195a.—Sullivan Room-and-Pillar Chain Machine, Class CE-7.

here, had the same general lines as the chain machine in Fig. 194. The sliding cutter head carried a transverse bar, driven by a sprocket chain and in which teeth were set. Length of cutter bar, 3-3½ ft., speed of rotation, 200 revs. per min., depth of cut, 4-5½ ft.

TABLE XXXIX

GENERAL DIMENSIONS OF SULLIVAN AIR-DRIVEN COAL CUTTERS

	Class CE 7 Room Work	Class CH 8. Longwall
Air pressure, lbs	20-70	20-70
Horse power of motor	30	30
Cutter bar, length, ins	66, 78 or 90	24, 30, 36, 42, 48, 54, 60 or 66
Distance, face to props, ins	70	30
Height when cutting, ins	24½	18¼
Height on standard truck, ins	34¼	28¼
Height on drop-axle truck, ins.	30¼	24¾
Height of groove cut, ins	5¼	5
Feed along face, ins per min	15-30	12-30
Weight, machine only, lbs	5800	5300
Weight, truck only, lbs	1000	950
Weight, equipment only, lbs	850	850

Fig. 196 shows a recent British bar cutter of a wholly different design, built for longwall mining by Mavor & Coulson, Glasgow, Scotland. It may be mounted on runners, sliding on the floor, or on wheels for travelling on a track. By mounting it on skids, as in the lower part of the illustration, the cut can be made in any part of the seam, between floor and roof. The machine will work in seams up to a pitch of about 60°. Fig. 197 shows the mode of operation.

The cutter bar, which is driven through gearing by a double-cylinder air engine, or an electric motor, can be swung through a total horizontal arc of somewhat more than 180°. The bar is threaded spirally, to remove the cuttings from the groove, and staggered bits are set in sockets throughout its length.

This machine is made in 3 sizes, ranging from 7 ft. 9 ins. to 11 ft. long (without the cutter bar), 1 ft. 4 ins. to 2 ft. high,

and 3 ft. 4 ins. to 3 ft. 11 ins. wide; shipping weight, 3,640 to 6,160 lbs.; maximum depth of undercut, $3\frac{1}{2}$–6 ft.; air pressure at the machine, 45 lbs.

FIG. 196.—Rotary-Bar Coal Cutter, Mavor & Coulson

Disk Cutters are a very old type, invented in Great Britain. A machine formerly made by the Jeffrey Manufacturing Co. is shown in Fig. 198. Disk cutters are still employed to some

extent in Europe, but are almost obsolete in the United States. They are designed for longwall mining only, and are especially useful in very thin, steeply pitching seams, being fed along the

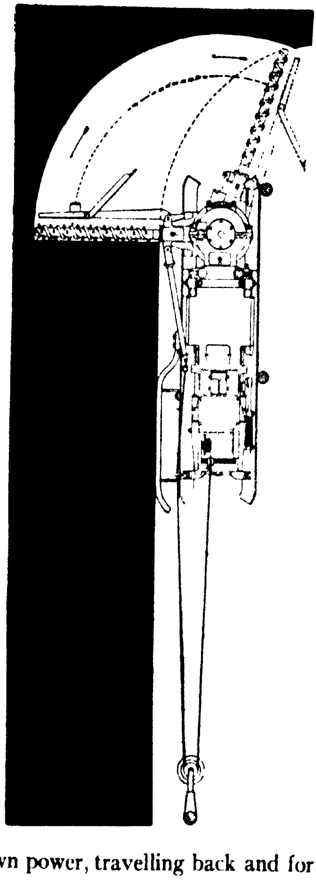


FIG. 197.—Mode of Operating the Mavor & Coulson Bar Cutter.

face by their own power, travelling back and forth, up and down the dip.

The cutter disk is supported by a bracket on one side of the frame, and projects a distance nearly equal to its own diameter. On the periphery is a series of staggered bits, cutting a groove

Fig. 198.—Jeffrey Disk Coal Cutter, Style 22-C, Compressed-Air Driven

high enough to admit the disk freely. The speed of the wheel is usually 15–30 revs. per min., depending on the character of the coal. When in operation the machine is automatically pulled along the face of coal by a wire rope, made fast to a post at the end of the face, and wound in by a drum geared to the driving engine.

A recent British disk cutter, built by Mavor & Coulson, Glasgow, is driven by electricity only.

Pick Machines in general construction resemble the reciprocating rock drills; but all work without rotation of the bit,

FIG. 199.—Sullivan Coal Pick, Working in a Thin Vein.

since in undercutting there is no question of preventing rifling of the hole, as in rock-drilling. Fig. 199 shows a typical case of a pick machine in operation. The machine is mounted on a wooden platform, 3 ft. wide by 8 ft. long, which slopes towards the face of coal, at an angle of about 5°. The recoil of the blows is thus nearly neutralized by gravity and the machine kept up to its work The operator chocks the wheels with wooden blocks, sometimes strapped to his boots, and directs the blows by swinging the machine laterally, with the wheels as a fulcrum. To give the machine sufficient reach, the front

cylinder head and piston rod are very long. A horizontal width of 4 or 5 ft. of undercut is thus readily commanded. Depth of cut rarely exceeds 5 ft. A helper clears away the débris with a long-handle shovel, and assists in moving and setting up the machine.

Most pick machines run at 200–250 strokes per min. The lower speed machines probably have some advantage, because, as each individual blow is directed by the operator, he can increase the efficiency of the work if he has time between strokes to point the pick in such a manner that it will do most execution. In average coal, an undercut of 4 by 4 ft. in horizontal area can be made in 16–18 mins. The platform can be shifted sidewise to the next position and the bit changed in 8–10 mins. Height of undercut is 12–14 ins. at the face, tapering to 3 or 3½ ins. at the bottom. Under favorable conditions, good operators can undercut, per shift, 75–85 linear ft. of face, to a depth of 4–4½ ft.; fair, average work, 60–65 ft. of undercut, 4 ft. deep.

The Harrison Pick machine, one of the oldest of this class, was invented in 1877. It has been built for many years by the George D. Whitcomb Co., Rochelle, Ill. Fig. 200 shows the longitudinal section. The valve is a long, double spool, actuated through a crank and connecting rod by a horizontal rotary engine set above the valve chest. The main cylinder of the machine has double ports at each end to cushion the stroke, and for running with a short stroke when desired.

These machines are made in heavy and light patterns, weighing respectively about 700 and 500 lbs. The heavier pick will cut to a depth of 5 ft. and is especially adapted to "shearing"; that is, making a vertical cut or groove, on one or both sides, in driving entries. For this purpose, the supporting wheels are 34 or 40 ins. diameter, to raise the machine high enough to give the requisite reach. Smaller wheels are used for ordinary undercutting.

The Sullivan Pick (Fig. 201) is made in 3 sizes, with cylinder bores of 4½, 4¾ and 5⅛ ins., for undercutting to maximum depths of 4½–6 ft. Weights, 650–850 lbs. Air consumption: 4½-in. machine, about 110 cu.ft. free air per min.; 5⅛-in. machine,

130 cu.ft. Standard wheels, 12–24 ins. diameter; wheels for shearing, 26–48 ins. Fig. 199 shows the pick at work. The spool-valve (123) throws the long flat valve (126), which controls the main ports (148, 149) and the secondary ports between them. These secondary ports produce cushioning at each end of the stroke, as follows: A check-valve (130) is inserted in the forward main port (149), and, when the pick does not strike the coal, the piston runs forward far enough to form an air cushion in the front end of the cylinder. This closes the check-valve, and prevents immediate admission of live air for the return stroke, which is begun by the cushion air.

In the hollow piston is a rifled-nut (104), engaging a rifle-bar (105). As the forward end of the piston rod is nearly square in section, and therefore cannot rotate,

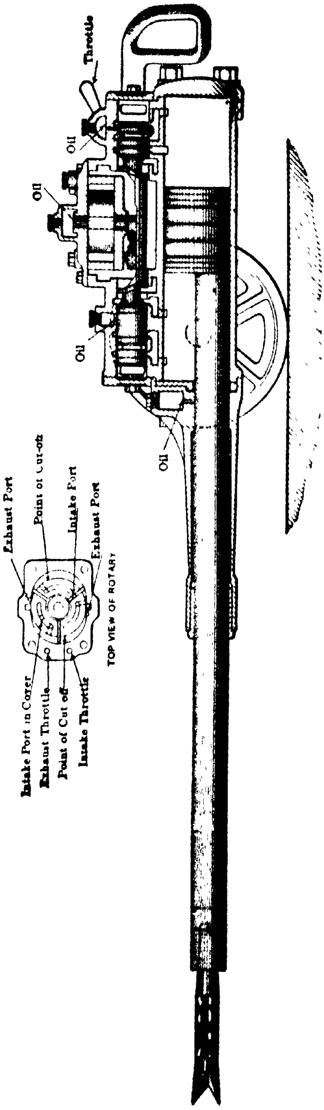
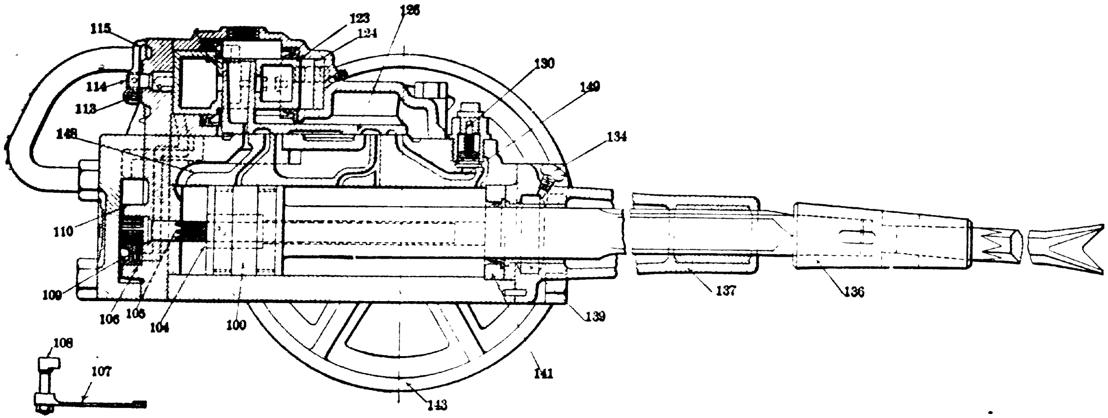


FIG. 200.—Harrison Pick Machine (Types PG and PW).



100. Piston (bare).
104. Rifle-nut.
105. Rifle-bar, with gear.
106. Seat for 109.
107. Spring pointer for 108.
108. Stem for adjusting 106.
109. Reverse valve.
110. Valve plate.
113. Spring for 115.
114. Regulating valve.
115. Index lever for 114.
123. Valve (piston).
124. Buffer for 123.
126. Valve (flat).
130. Check-valve.
134. Plug for oil hole.
136. Chuck.
137. Head (front).
139. Bushing in 137.

FIG. 201.—Sullivan Coal Pick.

the rifle-bar is rotated at each stroke; and the small gear on the back end of the rifle-bar rotates the reverse valve (106, 109). This valve in turn operates the spool-valve (123), by the dotted ports in the rear end of the cylinder. By a regulating valve (114), the operator adjusts the speed of stroke as the working conditions require.

Ingersoll-Rand Pick (Fig. 202). The valve motion is shown diagrammatically by Fig. 203.

Main ports S, S^1 are controlled by slide-valve G, on the back of which is a lug H, engaging the spool-valve F. Air enters alternately the opposite ends of F through ports J, J^1, which are controlled by the auxiliary slide-valve K and its spool-valve F^1. Valve F^1 is operated by the ports N, N^1, N^2 and N^3, which connect on either side with main ports S, S^1. Thus the rear end of the chest of F^1 is connected with forward main port S and the forward end with the rear main port S^1; hence, when air is admitted to the cylinder O through port S^1, a small portion of it passes through N, N^3 and throws valves F^1 and K. This admits air from F through port J^1 to spool-valve F and reverses

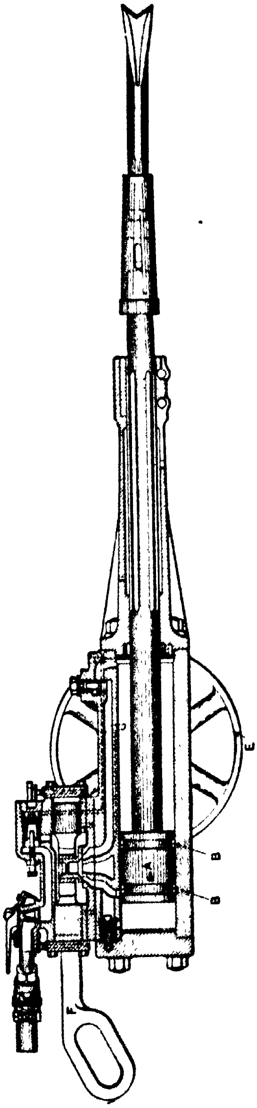


Fig. 202.—Ingersoll-Rand Coal Pick.

valve G, thus opening S to live air and S^1 to the exhaust. Some air passes from S through ports N^1, N^2 to valves F^1 and K, by which first K and then G are thrown forward, reversing the main ports and completing the cycle. The speed of F^1 and K, and hence of F and G, is regulated by the screws L, L^1.

When the piston passes the double port P the exhaust ceases. Therefore, if the pick misses the coal, the piston on advancing beyond P cushions on air in the forward end of the cylinder.

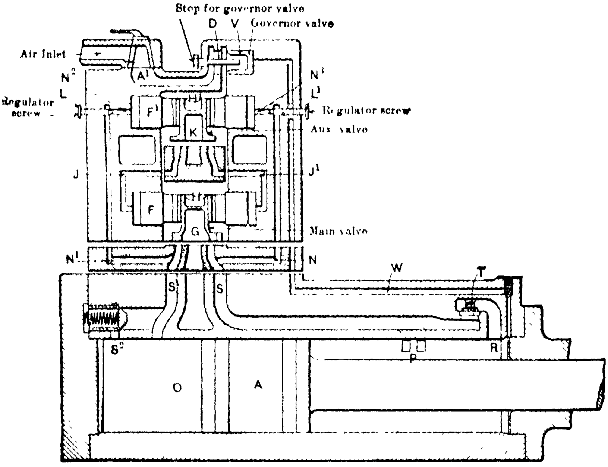


FIG. 203.—Ingersoll Rand Coal Pick. Diagram of Valves and Ports.

This high-pressure air forces back the governor valve V, wholly or partly cutting off the inlet air to valve K, so that the machine runs at reduced speed until the bit again strikes the coal before port P is covered. The regular exhaust then takes place, the governor valve V opens, and the machine resumes regular operation. The throw of V is adjusted as desired by the stop D. The back stroke is begun by the cushion air confined by the check-valve T. No live air can pass port S until the piston has advanced far enough to reduce the cylinder pressure below that of the live air in S, plus the resistance of the check-valve spring. The back stroke is cushioned by ports S^1 and valve S^2.

This pick is made in 3 sizes: $4\frac{1}{2}$, 5, and 6 ins. diameter. Maximum depth of cut, 4–6 ft.; gross weight of machine, 550–950 lbs. Standard wheels are 14 and 17 ins. diameter; larger wheels are used for shearing.

The Hardy "coal puncher," made by the Hardy Patent Pick Co., Sheffield, England, is a small, light machine, designed especially for driving headings, though it may be used also for room and longwall work. For swinging the machine on its mounting (usually a column), it has a worm gear, similar to that of the "Radialaxe" coal cutter (Fig. 204).

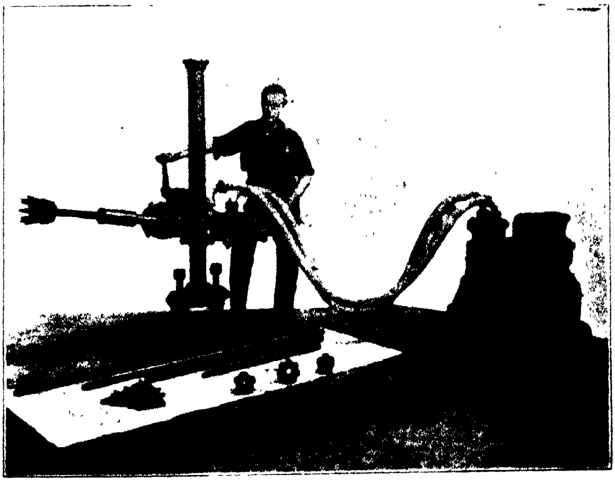


FIG. 204.—Ingersoll Rand "Radialaxe" Pick

Ingersoll-Rand "Radialaxe" Pick (Fig. 204) is intended for shearing in driving entries, and for mining in steeply pitching seams where chain machines or ordinary coal picks are not applicable. It is an adaptation of the Temple-Ingersoll Air-Electric drill (Chap. XX), mounted on a column, and provided with a worm and wormwheel sector. A handwheel on the worm spindle enables the operator to swing the entire machine while at work, in either a horizontal or vertical arc.

The group of bits, as shown, is set in a rosette socket, held by friction on the tapering end of the drill shank, which is similarly inserted in the deep socket of a long chuck. The individual bits are thus readily removed for sharpening and replacement.

Sullivan " Post Puncher " (Fig. 205) is a modified rock-drill, designed for the same service as the Ingersoll-Rand " Radialaxe." In ordinary cuts to depths of 6–8 ft., when the column cannot be set up close to the face, the reach of the machine is increased by using an extension shank.

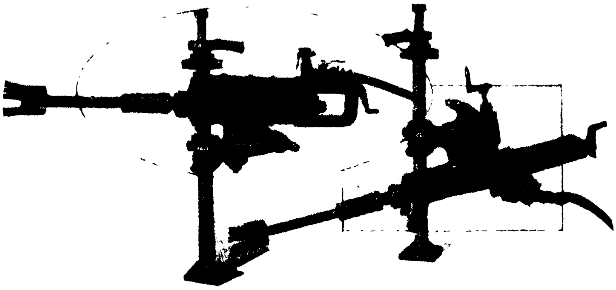


FIG. 205.—Sullivan " Post Puncher " At the left, the machine is arranged for breastwork or undercutting, at the right, for shearing

The bit is either solid, or of the rosette type, comprising a group of 5 or 7 independent, removable bits. By a segment and worm on the arm of the column mounting the machine is swung as the work requires. The cylinder is $3\frac{1}{2}$ ins. diameter; weight of machine, 226 lbs.

Pneumelectric Coal Puncher, as its name implies, uses both compressed air and electricity. A small electric motor drives a pair of independent pistons in a cylinder (Figs. 206, 207). The rotary motion of the motor is changed to the rectilinear motion of the piston by the following device: The driving pinion *B* engages the horizontal gear *C*, which has a solid web, carrying a stud *D* (Figs. 206, 208). On *D* is a gear *E*, and a crank with crank-pin *G*. Within the main gear *C*, and attached rigidly to it, is a gear *F*, with internal teeth engaging the crank-pinion *E*;

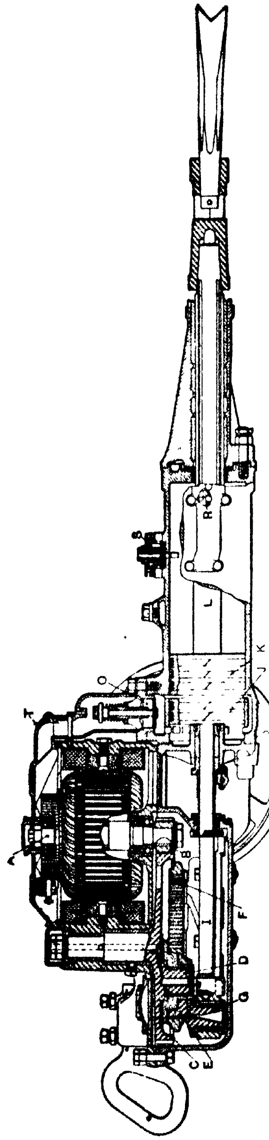


FIG. 206.—Pneumelectric Coal Puncher.

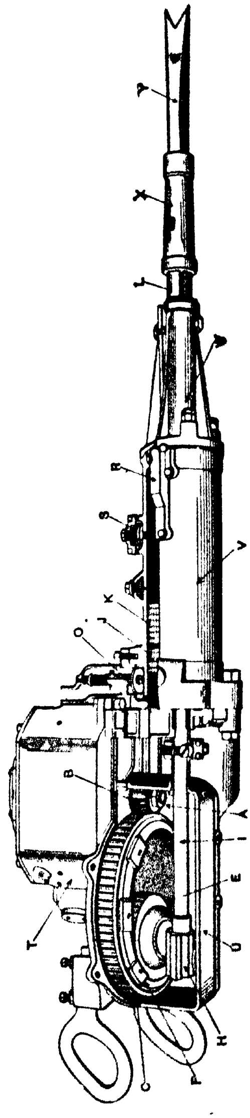


FIG. 207.—Pneumelectric Coal Puncher.

F having 66 and *E* 33 teeth. These gears are so proportioned that stud *D* revolves in a circle concentric with *F* and of one-half its pitch diameter. Gear *E*, revolving freely on *D*, causes crankpin *G* and crosshead *H* to reciprocate between the guides. To the crosshead is attached the piston rod *I*, with its piston *J*.

The cylinder *V* contains two pistons, entirely unconnected with each other: the rear or driving piston *J*, and the forward piston *K*, with its rod *L* and chuck *X*. In starting the machine on its first forward stroke, piston *J* simply pushes *K* forward, air meantime entering the cylinder behind the piston, through

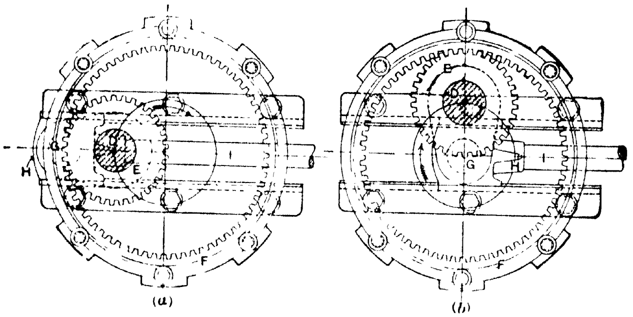


FIG. 208 Pneumelectric Coal Puncher Diagram of Gearing.

the valve *O*. On the back stroke the air in the rear of the cylinder is compressed and the air between the pistons is rarefied, thus causing *K* also to make its return stroke by suction, while air enters freely through a port at *R*. At the end of the back stroke, the air passages below valve *O* lie between the pistons, whereby the charge of compressed air enters the cylinder and drives piston *K* forward on its first regular stroke. The stroke is cushioned after *K* passes the port at *R*. Piston *K*, having completed its forward stroke, is followed by piston *J*, the air between them being exhausted through valve *S*. The return stroke is then made by both pistons, as at first.

The diameter of the cylinder is 6¼ ins., and the rear end clearance spaces are proportioned to produce a working pressure of 95–100 lbs. The motor is designed to run at three speeds,

under the operator's control, giving to the pick 140, 160 or 180 strokes per min. It is stated that 7 H.P. are required to run the machine.

The Stanley Header, originally brought out in England, is intended for development work in collieries, driving circular headings, for entries, airways, etc. It is now rarely used. In one of its forms (Fig. 209) a crosshead *a*, mounted on a screw shaft *b*, and carrying two horizontal arms *c*, cuts an annular groove, 3-4 ins. wide. A central core is thus left, which either breaks up and is shoveled back as the work advances, or is blasted or

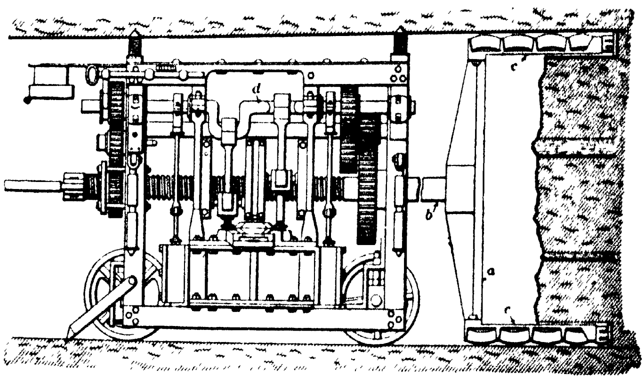


FIG. 209 - Stanley Heading Machine for Collieries.

wedged down from time to time, if necessary. The screw shaft is driven through gearing by a pair of compressed-air cylinders. Differential gearing produces the feed. The whole is carried in a frame on wheels, held firmly when in operation by jack-screws set against the roof. The machine is narrow enough to permit a man to pass alongside to the front, to throw back the broken coal and keep the cutter head free.

Modifications have been introduced in this country and abroad. In one of them, the entire section of the heading is taken out in a single operation. For this, the crosshead and arms are replaced by a flat, cone-shaped head, carrying a number of individual bits, arranged in diametral lines on the cone.

The circular paths traversed by the bits cover one another, so that the whole mass of coal is broken up. This modification has been used in western Pennsylvania, in some of the Frick Coal and Coke Co.'s mines. Average speed of advance, under favorable conditions, $2\frac{1}{2}$–3 ft. per hour, including moving and setting up. In one case 2,254 linear ft. of entry were driven at an average speed of 17 ft. per 9 hours, and an average working cost of 40 cents per ft. The exhaust assists in ventilating long headings.

For rapid development work in longwall mining, the Stanley Header has been used by the Colorado Fuel Co., with the following results:

HAND LABOR

2 men, 1 10-hr shift	$ 4 00
Paid to men for coal produced in driving, $4\frac{1}{2}$ tons at 50 c	2 25
Cost	$6 25
Distance driven in 10 hours, 3 ft	

MACHINE WORK

1 operator, $3 00, 1 helper, $2 50	$ 5 50
3 shovellers to load coal behind machine at $2 00	6 00
Compressed air, repairs, depreciation and interest	3 50
Squaring up corners, for timbering and track	5 00
Cost	$20 00
Distance driven in 10 hours, 20 ft.	

Crediting to the machine work the coal produced, *viz.*, $15\frac{1}{2}$ tons at 50 cents loaded, the net cost for 20 ft. of entry was $1.84 per yard.

Auger Drills, for boring holes in coal, rock-salt and other soft material, are operated by compressed air or electricity. The Ingersoll-Rand Co. makes a breast drill, resembling a machinist's breast drill. It has a 3-cylinder motor, which can readily be reversed for withdrawing the bit from the hole; weight, exclusive of the bit, 18 lbs. Another heavier machine is mounted

on a column or bar. A compressed air auger mounted on single or double column, is made by the Jeffrey Manufacturing Co.; weight, for a 6-ft. vein, 183 lbs. Speed of the engine is about 3,000 revs., and that of the feed shaft, 850 revs. per min.

Auger drills operated by compressed air, electricity or hand power, are also made by the Howells Mining Drill Co., Plymouth, Pa.

The Fairmont Mining Machinery Co., Fairmont, West Va., makes an auger drill for mounting on electric-driven chain coal cutters.

Comparison of Coal Cutters. The chain machines, which have the widest application, work best in clear coal, of uniform quality, though the recent types are suitable also for hard, "bony" coal, or coal containing streaks of pyrites ("sulphur balls"). In coal of this character, the pick machines work well, because the operator can regulate the strength of the blow and so direct the machine as to cut around a hard place.

Somewhat less slack and fines are made by the chain machines, as the volume of undercut is smaller; but, when the product goes to coke ovens, the larger quantity of fines made by pick machines is immaterial. Also, in solid, hard coal, the higher undercut of the pick machines causes a more complete breaking up of the whole mass when blasted down, and the coal, therefore, is sometimes more readily loaded.

For chain cutters a fairly good roof is desirable; otherwise props may have to be set so close to the face as to interfere with the manipulation and shifting of the machines. Chain machines can be worked in seams as thin as about 20 ins., though they are more conveniently operated in thicker seams. The continuous-feed chain and disk machines are specially useful for thin, pitching seams in longwall work, as they operate with almost equal facility either up or down the pitch. Disk cutters are now rarely used.

The "mining rate," or cost of mining by hand, together with the character of the coal seam, will usually determine whether coal cutters can be economically applied in a given mine or district. In general, for seams of average quality and thickness,

FIG. 210.—Jeffrey Air Loader for Collieries (Class 38-A)

when the local cost of hand mining is not less than 55–60 cents per ton, a saving may be effected by introducing machines.*

Loading Machines of several makes are used underground to some extent for loading cars in the working places of both coal and metal mines. Fig. 210 shows a compressed-air driven loader of the conveyer type, for collieries, which may be mentioned here in connection with coal-cutting machinery. A considerable number of them are now employed in the United States and Canada. The frame of the motor supports the elevated end of the conveyer, near the car.

* This applies to pre-War conditions.

CHAPTER XXIII

CHANNELING MACHINES

ORIGINALLY, channeling machines were used almost exclusively for getting out dimension stones in quarry work. Of late years, however, they have been employed in increasing numbers for certain kinds of rock excavation, where it is desired to have smooth, uniform walls; for example, rock cuttings for railroads, canals and water-wheel pits for power plants. They are best adapted for cutting the softer rocks, like limestone, most of the sandstones, slate, shale, etc., though they may be used also for some of the varieties of granite, gneiss, porphyry, schist, and other metamorphic rocks. Hard rocks are best quarried by drilling rows of holes, with wedging or blasting.

In a certain sense channelers resemble reciprocating rock-drills, a single bit or a "gang" of bits being attached to the piston rod. But, instead of drilling a series of round holes, the channeler, as its name implies, cuts a continuous, narrow groove, without rotation of the piston and bit. In its typical form, the machine is solidly supported on a heavy carriage or truck, generally mounted on a track laid along the line to be cut. The motive power may be compressed air, electricity, or steam. By means of an auxiliary engine and worm gearing, the whole machine, while at work, is fed forward automatically at a suitable speed. Fig. 211 shows the general construction of a standard compressed air-driven channeler. Another design, a track channeler for cutting marble, with adjustable mounting and a re-heater mounted on the carriage, is shown in Fig. 212.

General Construction. The construction of channeling machines is varied to suit the conditions of work: 1. For cutting vertical channels only, the rigid head machine is used; that is, the standard supporting the cylinder and accessories is non-adjust-

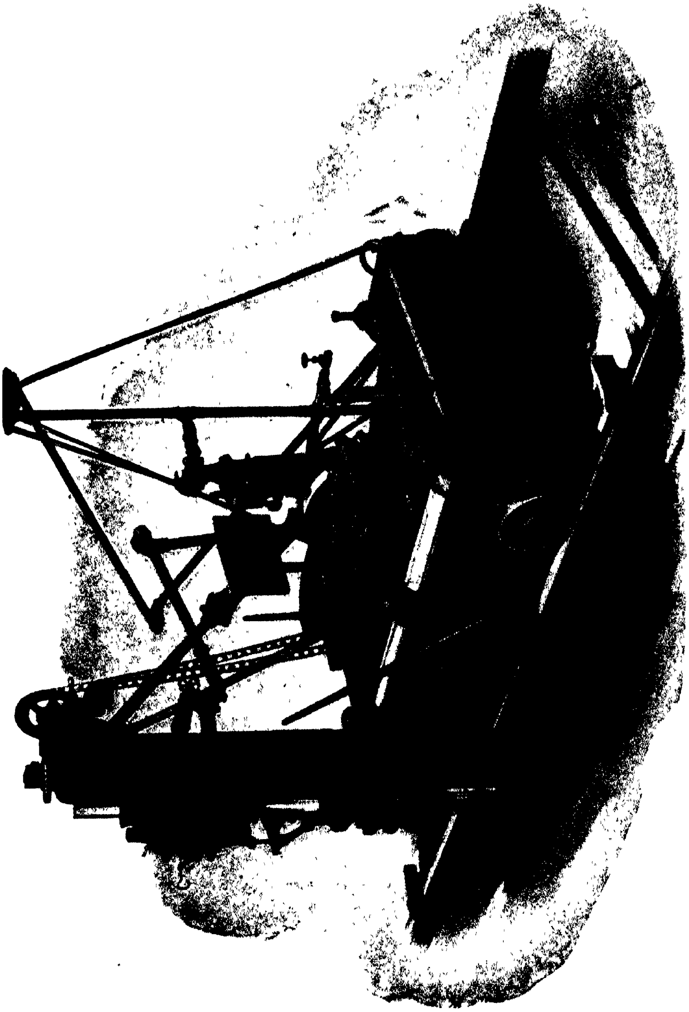


FIG. [illegible].—Sullivan Track Channeler.

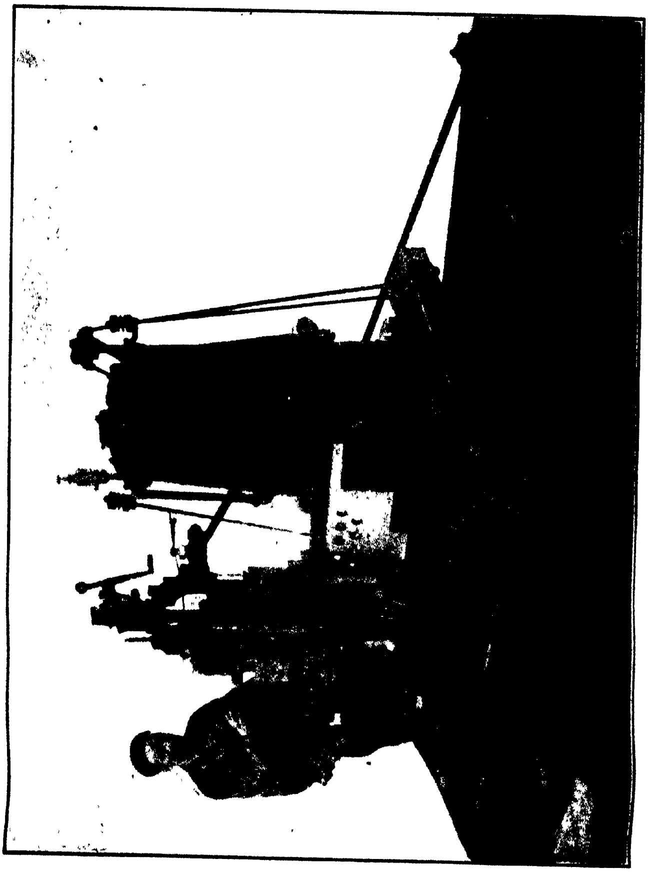


FIG. 212.—Ingersoll-Rand Ram Track Channeler, for Marble

able, being permanently fixed in an upright position. Fig. 213 shows a steam-driven channeler of this class. It is employed for canal and railroad cuttings, general rock excavation and for

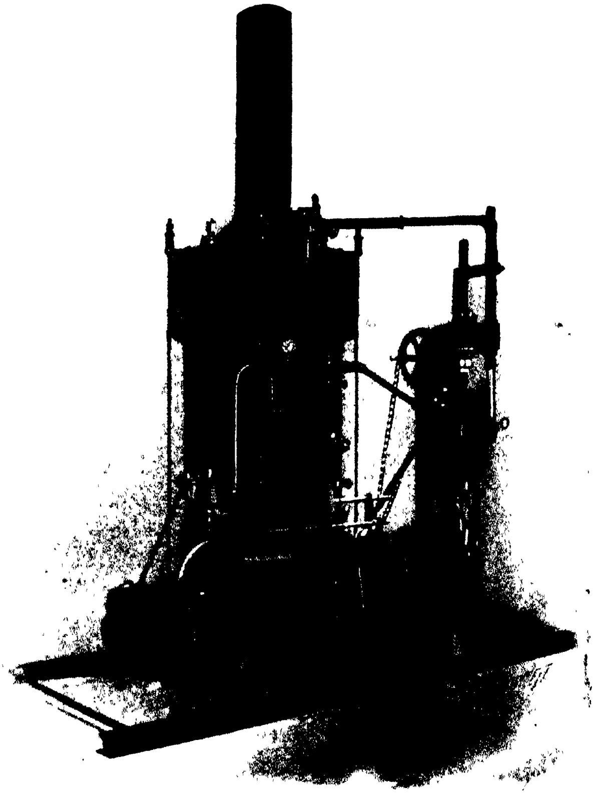


FIG. 213.—Sullivan Rigid Back, Steam-Driven Channeler.

quarrying where the strata are horizontal or nearly so. 2. For quarrying building stone lying in inclined stratified beds, like most limestones, the channels must generally be cut at right angles to the bedding planes; hence, the mounting of the cutting

engine is adjustable, for making a channel at any desired angle to the vertical. The cylinder with its appurtenances is swivelled on its supporting frame or standard; or the frame may be provided with T-slots (resembling those of the table of a planer), by means of which the cylinder is bolted firmly in the required position. The supporting frame, in turn (see Fig. 212, of the Ingersoll-Rand Ram Track Channeler), may be swung back to a nearly horizontal position, for making wall cuts along a quarry face. In different designs, the minimum swing-back angle varies from 15° or 20° to 33° to the horizontal; but it is not often necessary to cut with these machines at less than 45°. Fig. 214 shows a Sullivan channeler of this class. 3. A third form, designed specially for making horizontal channels, or "under-cuts," is used less frequently than the others. An Ingersoll-Rand machine of this type is shown in Fig. 215. As indicated in the cut, the head may be bolted to either end of the carriage. 4. Lastly, a light machine, which is in effect a large rock-drill, may be mounted on a "quarry bar"—a long, hollow bar, supported at each end by a pair of inclined legs.* This is generally used for drilling a row of holes placed close together, the partitions between which are afterward cut out by a "broaching" bit. A modification of the quarry-bar machine is made by the Ingersoll-Rand Co. It is a true channeler, mounted on a heavy swivel plate, which slides on a pair of horizontal bars, about 10 ft. long, supported by inclined legs (Fig. 216). The whole machine is fed along the bars automatically by a small, 3-cylinder engine, which actuates a travelling feed-nut, engaging with a threaded shaft between the bars.

There are many variations in construction of the above-mentioned classes of channeler, to adapt them to local conditions. Among other machines, of entirely different design, may be mentioned the Wardwell and the Bryant channelers. The Wardwell, a heavy machine operated by steam only, has been in successful use for many years. It is intended for making vertical channels, a gang of bits being set in a

* In one of the Sullivan models, the legs are replaced by vertical standards, each carried on a small wheeled truck.

massive frame, which is given an up and down movement, something like a jumper drill.

For the first three classes of channeler a gang of from 3–5 bits is employed. These have long square shanks and are set closely side by side, the cutting edges being alternately at right

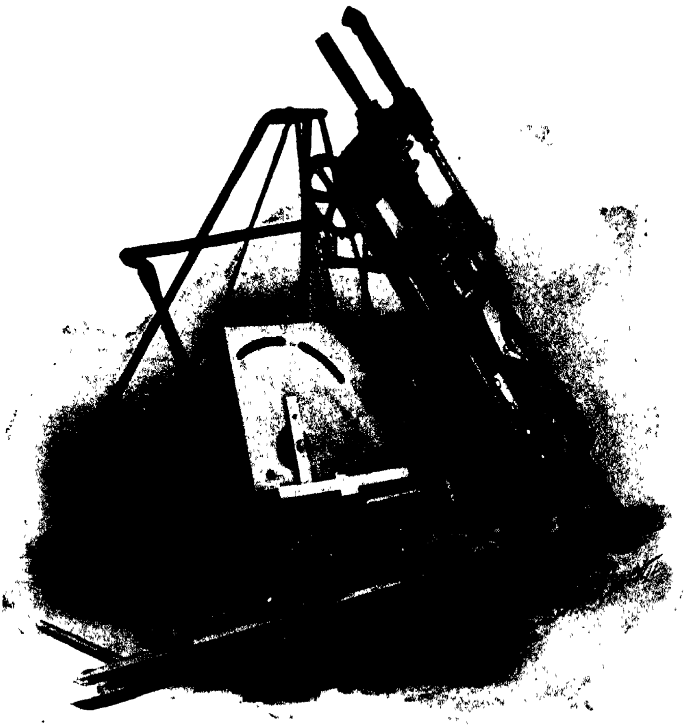


FIG. 214.—Sullivan Adjustable Back Air-Driven Channeler.

angles and at 45° to the direction of the channel. This arrangement forms practically a succession of Z-shaped bits, and insures the cutting of a regular channel with smooth walls. The bits are clamped firmly in a heavy chuck, attached to the piston rod of the engine, and are guided either by a crosshead or (as in some

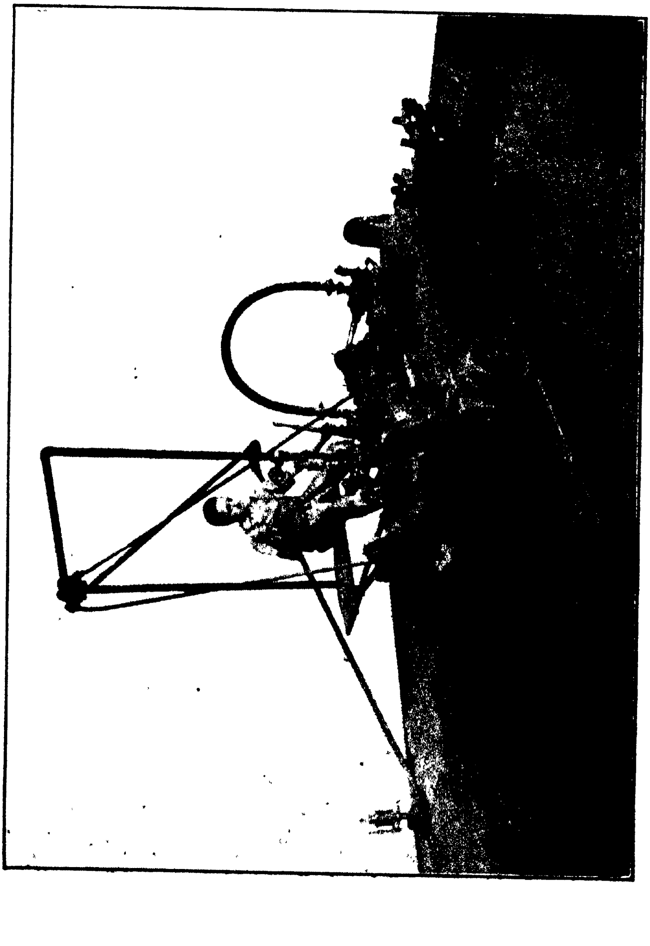


Fig. 215.—Ingersoll-Rand Undercutting Track Channeler. Type HF-3.

FIG. 216.—Ingersoll-Rand "Broncho" Channeler.

of the Ingersoll-Rand patterns) by a pair of roller guides. Some saving in power has been realized by the introduction of the roller guides; they eliminate part of the weight due to the crosshead, which must be lifted at each stroke, and the friction loss is reduced. The Sullivan Company builds a duplex or double-head machine. There are two cylinders, side by side on a heavy frame, each with its gang of bits and operated by a single valve-chest. As the blows alternate, one piston making its down stroke while the other is on the up stroke, the machine can be run at high speed without excessive vibration. Its working capacity is correspondingly greater than that of the single-cylinder machines. When the plant consists of a few machines only, they may be advantageously driven by steam (Fig. 213); but, for large-scale work, a higher degree of economy results from the employment of compressed air, furnished by a central plant. Each machine is then provided with its own reheater,* mounted on the carriage (Fig. 212). Air pressures generally range from 85–110 lbs.

While at work the main cylinder of the channeler is raised and lowered in its guide shell by a screw-feed, operated automatically or by hand. The hand feed is rarely employed except for the smaller machines. The automatic feed may be caused either by an independent engine, similar to that used for the longitudinal feed of the quarry-bar machine, already referred to; or, by a chain and sprocket drive from the machine which furnishes the propelling power along the track. The chain feed, as used in the Sullivan channelers, is shown in Figs. 211 and 213. Most of the Ingersoll-Rand channelers are provided with the independent feed engine, which is of the 3-cylinder type, very small and compact in design. In either case, when the cut has reached the required depth, the feed is reversed and the entire head, with its accompanying parts, is raised preparatory to making the next cut.

Depth of Cut and Speed of Work. The heaviest channelers—those with rigid back or standard—will cut to depths of from 8

* See Chap. XIX, on Reheaters.

–15 or 16 ft., according to the character of the stone; the swing-back and bar machines will cut from say 6–10 or 12 ft., and undercutting machines up to 7 ft. For starting a channel, the width of a bit is from $1\frac{1}{2}$ to a maximum of 4 ins., depending on the depth of cut to be made and on the nature of the stone. The gages of the successive bits are generally reduced by $\frac{1}{16}$ in. each, the finishing bits usually cutting a width of $1\frac{1}{8}$ in.

The cutting capacity of channelers varies greatly. It is largest in the softer stones, when of uniform texture and quality, and in fully developed quarries, where the work is systematic and the stone lies below the zone of weathering and surface disintegration. In sandstone of average hardness and under favorable conditions, from 250–300 sq. ft. of channel may be cut per 10 hours by the heavy machines; or, including all stoppages and delays, from 4,000–4,500 sq. ft. per month; in the softer sandstones and limestones higher duties are obtainable. The swivel-head and other adjustable channelers are lighter than the fixed-back machines and in the same kind of stones their rate of work is generally slower. Machines working in rather hard marbles, like those of Rutland, Vt., will cut from 2,300–2,500 sq. ft. per month, or an average of 85–100 sq. ft. per day. A single day's work, however, will often greatly exceed these figures. In hard marble or limestone, the smaller bar machines will cut an average of say 40 sq. ft. per 10 hours and up to 125 sq. ft. in softer stones. For hard gneiss, or schist, like that of New York island, an average duty would be 65–70 ft. per day.

Tables XL and XLI, showing dimensions, weights, and other data, of the channelers of two well-known builders, will further illustrate the features of these machines.

The Ingersoll-Rand Company have applied the principle of their "Air-Electric" rock-drill to the design of the cylinder and air compressing mechanism of a track channeler (Fig. 217). That is, an electric motor, mounted on a carriage, drives a single-acting air compressing pulsator, which is connected to the channeler cylinder in a manner similar to that of the Temple-

TABLE XL

SPECIFICATIONS OF INGERSOLL-SERGEANT CHANNELERS

Size and Type.			Fixed Back Channeler		Swing Back Channeler			Under-cutting Channeler	Broncho Channeler.
			H8	H9	H9	6 in	5 in		
Diameter of cylinder		in	8	7	7	6	5	$3\frac{1}{2}$	$3\frac{1}{2}$
Length of stroke		in	9	9	9	$6\frac{1}{2}$	$6\frac{1}{2}$	7	$6\frac{1}{2}$
Distance of cut from vertical wall		in	$7\frac{1}{4}$	$7\frac{1}{4}$	$7\frac{1}{4}$	$7\frac{1}{4}$	5	$8\frac{1}{2}$ (lift)	
Distance from center to center of cut with machine reversed		ft in	7-0	$6-0\frac{5}{8}$	$6-8\frac{5}{8}$	$4-7\frac{5}{8}$	$4-6\frac{5}{8}$		
Inside gage of track		ft in	5-3	$4-4\frac{5}{8}$	$4-4\frac{5}{8}$	$3-0\frac{5}{8}$	$3-0\frac{5}{8}$		
Length over all		ft.in	5-3	5-3	5-2	5-5	5-5	$4-0\frac{3}{8}$	
Width over all	Without boiler	ft in	7-1	$7-0\frac{1}{2}$	$7-4\frac{1}{2}$	5-5	5-2	$5-10\frac{1}{2}$	14-0
	With boiler	ft in	$7-6\frac{1}{2}$	7-6	7-10			8-3	2-6
	With reheater	ft.in.	$7-3\frac{1}{2}$	7-3	7-7				
Height*	Without boiler	ft in	7-4	7-4	7-2	$6-10\frac{1}{2}$	$6-10\frac{1}{2}$	2-10	6-0
	With boiler	ft.in	10-0	10-0	10-2				
	With reheater	ft in	7-4	7-4	7-2				
Weight of Channeler alone	Without boiler	lbs	9,000	9,000	8,000	5,150	4,850	6,800	2,375
	With boiler	lbs	12,000	12,900	11,000				
	With reheater	lbs	10,300	10,300	9,300				
Total shipping weight with track and equipment	Without boiler	lbs	†13,900	†13,900	13,700	†10,500	†10,200	†11,800	3,500
	With boiler	lbs	†17,875	†17,875	17,675				
	With reheater	lbs	†15,175	†15,175	15,000				

* Height is from top of rail to top of boiler hood, which does not include stack
† These weights are for domestic shipment. Add 1 000 lbs for foreign shipment.

TABLE XLI

SULLIVAN STONE CHANNELERS—SIZES, SPECIFICATIONS AND WEIGHTS

Size	Type.	Diameter of Cylinder Ins	* Height of Machine ft	ins	Length Along Track ft	ins	Width ft	ins	Gage of Track, Inside Measurement ft	ins	Distance, Center of Cut to Wall Ins	Distance between Centers of Cuts when Machine is Turned on Track. ft	ins	Free Air Consumption at 100 Lbs. Pressure Cu ft per Min †	Weight in Lbs. Machine only.	Weight in Lbs. Equipment only	Weight in Lbs. Total.
Y-8	Rigid head with boiler	8	9	11	6	6	6	10	4	11	7	6	8		13,270	6,050	19,320
Y-8	Rigid head without boiler	8	8	3	6	6	6	10	4	11	7	6	8	400	‡13,900	5,830	19,730
Y-8	Double head without boiler	8	8	3	6	6	6	10	4	11	7	6	8	750	16,500	6,800	23,300
Y	Rigid head with boiler	7	9	11	6	6	6	9	4	11	6	6	6		12,235	6,050	18,285
Y	Rigid head without boiler	7	7	6	6	6	6	9	4	11	6	6	6	300	‡12,860	5,830	18,690
Y	Double head without boiler	7	7	6	6	6	6	9	4	11	6	6	6	550	10,230	6,800	17,030
6½	§ Rigid head with boiler	6½	9	11	6	6	6	10	4	11	9	6	3		10,520	5,435	15,955
6½	§ Rigid head without boiler	6½	6	3	6	6	6	7	4	11	9	6	3	230	6,520	5,215	11,735
Z	Swivel head with boiler	7	9	11	6	6	6	9	4	11	6	6	6		13,000	6,050	19,050
Z	Swivel head without boiler	7	7	6	6	6	8	5	4	11	6	6	6	300	9,000	5,830	14,830
Z	Double head without boiler	7	7	6	6	6	6	5	4	11	6	6	6	550	11,210	6,800	18,010
VW	Swivel head duplex with boiler	6½	10	3	6	6	6	2	4	11	8	6	6		13,900	8,600	22,500
VW	Swivel head duplex without boiler	6½	6	7	6	6	6	7	4	11	8	6	6	500	8,900	8,575	17,475
6½	Swivel head with boiler	6½	9	11	6	6	6	9	4	11	6	6	6		11,000	5,430	16,430
6½	Swivel head without boiler	6½	6	8	6	6	6	5	4	11	6	6	6	230	7,200	5,215	12,415
6½	Double head without boiler	6½	6	8	6	6	6	5	4	11	6	6	6	410	8,900	6,080	14,980
VX	Swivel head without boiler: ‡ Standard on bar to cut from vertical to horizontal.	4½	5	10	4	3	4	5	3	3	4½			190	2,600	3,140	5,740
VX	Swivel head without boiler: ‡ Standard removed from bar and hung at either end of frame for undercutting	4½	2	6	variable				3	3	Height of bench 8½ in			190	2,600	3,340	5,940

* Machines with boiler are measured to the top of the smoke bonnet without the stack, machines without boiler, to the top of the standard, when vertical. † Without reheating. ‡ Includes three balance weights of 4,635 lbs. § These channelers are not equipped with the power hoist.

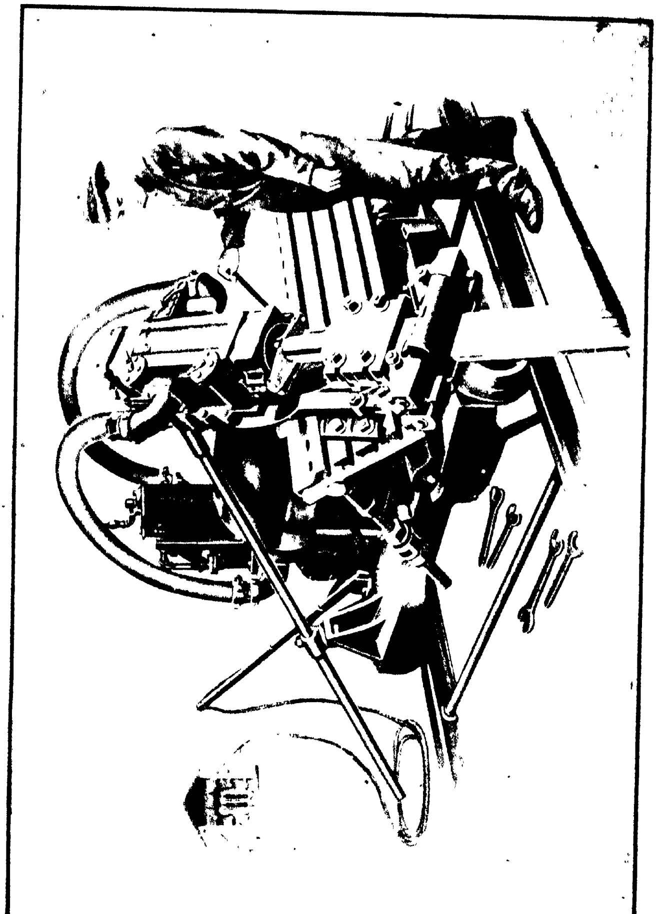


FIG. 217.—Gibson-Ingersoll "Electric-Air" Track Channeler, with Spring Rod.

Ingersoll rock-drill, described in Chap. XX. For each double stroke of the pulsator, there is a blow and return stroke of the channeler piston, the speed of stroke being thus controlled by the speed of the electric motor which furnishes the power. Favorable results, both as to power cost and maintenance, have been secured. This channeler has a swivel head and a swing-back support, and is therefore suitable for varied quarry service.

CHAPTER XXIV

OPERATION OF MINE PUMPS BY COMPRESSED AIR

It is intended here to deal only with that part of the extensive subject of mine drainage which has to do with the employment of compressed air as a motive power. Under this head there are three general forms of apparatus:

1. Direct-acting pumps: single-cylinder, duplex, or compound.
2. The air-lift pump.
3. Pneumatic displacement pumps.

In this chapter the first class only will be considered.

Simple, Direct-Acting Pumps. Notwithstanding the general similarity in the behavior of steam and compressed air, when used in the cylinders of direct-acting pumps, there are some important points of difference. By first considering briefly the construction of the types of pump in common use the results obtainable from the employment of compressed air can best be set forth.

The development of the direct-acting pump dates from Henry R. Worthington's invention in 1841; and a large part of all the pumping in the mines of this country, and much of it in other countries also, is done by pumps of this class. The cylinders are set tandem, the power being transmitted from the steam to the water cylinder through a piston-rod common to both. As there are no rotating parts, the length of stroke is controlled by the admission and exhaust of the steam. In all the simple pumps the valve motion involves the use of an auxiliary valve, the movements of which are governed by the reciprocating movement of the piston, and which in turn operates the main valve. The duplex form consists essentially of two simple pumps, set side by side, with an interdependent valve motion; that is, the valve of each is operated positively, through

a system of levers, by the movements of the piston of the other side.

Though direct-acting pumps are strong and reliable, simple in construction, and occupy but little space, they are extremely uneconomical machines, unless the steam cylinders are compounded. It is hardly necessary to say that this ought not to be the case. Pumping is an operation that should be conducted economically, especially in connection with mining, where the pumping of water is classed as "dead work." Moreover, the conditions in themselves are not unfavorable. A pump works under a practically constant load, from the beginning to the end of each stroke, the only necessary variation—which need not be large—occurring at the instant the discharge valves open.

The trouble is that, in attaining compactness, simplicity, and moderate first cost, the power is not applied in simple, direct-acting pumps to the best advantage. As there is a constant load, but no fly-wheel to equalize the power, steam must be admitted at full pressure throughout the entire stroke; otherwise the piston would be unable to reverse, and would come to a standstill. Such a pump must work practically without cutoff, and therefore a cylinderful of steam, nearly at initial pressure, is exhausted at each stroke. In some pumps the terminal pressure is quite as high as the initial. A duplex, non-compound pump, having a positive valve motion, may at times be even a more extravagant steam-consumer than a single-cylinder pump, since one piston may reach the end of its stroke before the other is ready to reverse its valve. In such case the momentum of the incoming steam fills the cylinder at initial pressure at the moment of exhaust.

For steam-driven pumps there are several ways of improving these conditions:

1. The adoption of compound or triple expansion cylinders. This type is suitable for the larger sizes of pump, and its use is increasing for mines where the depth and quantity of water warrant the higher first cost. The space occupied is little greater than for simple pumps of the same capacity, and satisfactory re-

sults are obtained when they work under proper conditions and with sufficient initial pressure.

2. While retaining the tandem form, a fly-wheel may be introduced, driven from the crosshead or from the steam-cylinder connecting-rod. This is a reversion to a type of pump long ago discarded for general service in this country, in favor of the simpler but less efficient form with no rotating parts. Although such a pump occupies much more space and its first cost is increased, there can be no doubt as to the advantages of being able to use the steam expansively, without the necessity of compounding. A large number of pumps of this description are now employed in mines; many of the Riedler pattern and some of less elaborate and expensive design, such as the Prescott and others, in which an early cutoff—at one-quarter or even one-eighth stroke—is satisfactorily adopted.

Notwithstanding the advances made along these lines in the mechanical engineering of pumps and the added economy gained in their operation, it has been very generally assumed in the past that similar economies are not attainable when compressed air instead of steam is employed as the motive power. Yet the advantages accruing from the utilization of compressed air transmission in mines are marked. As the heavy losses due to radiation and the condensation of steam in pipe-lines are avoided, the transmission of power by compressed air may be conducted with a high degree of efficiency. No difficulty exists as in the disposal of exhaust steam underground, nor is any danger to be apprehended from the rupture of a compressed-air pipe, while the bursting of a steam pipe in a shaft or in the mine workings may cause serious trouble. The failure to realize these advantages, and the unsatisfactory results obtained in most cases from compressed-air-driven pumps, are due largely to the fundamental differences in the behavior of steam and compressed air when used in a motor cylinder. In Chap. XVII reference has been made to the reduction of cylinder temperature accompanying the expansion of compressed air. The point of cutoff being the same, this causes lower terminal and mean pressures with air than with steam. In other words, at a given initial pressure

and without reheating, a cylinderful of air develops less power.

This property of air, together with the fact that it does not condense, indicates clearly that steam and compressed air are not equally well adapted for use in an engine of the same design. It is not easy to understand, therefore, why mechanical engineers and especially pump-builders have not given more attention to the production of pumps properly designed for the use of compressed air. Few, if any, other branches of motor-engine practice have been so neglected. Lack of information among users of compressed air is responsible in part; in addition to which it is not generally realized that relatively unimportant modifications, at small cost, would produce much better results. Users of the ordinary steam pump have become accustomed to its low economy, and, because it is strong and serviceable, it is apt to be accepted without question when compressed air is used instead of steam. But in applying compressed air to the inefficient single-cylinder pump, as usually designed for steam, the net result is no better and may be even worse, than that obtained from steam. The clearance spaces are large and, as the air is admitted to the cylinder throughout full stroke, it is used in a wasteful manner. Moreover, the stroke is often shortened by imperfections in the valve action.

Another unfavorable feature of mine pumps driven by compressed air is the frequently improper selection of the cylinder proportions and arrangement of the plant. In mines having a number of levels the pumps are distributed according to varying requirements as to height of lift and quantity of water to be raised. The lowermost pump may have to work under a heavy head; others under a head of only 100 or 200 ft. As all are usually operated from the same pipe-line and under a common air pressure, it is clear that the dissimilarity of working conditions must be met by proportioning the water and power ends of each pump according to the work to be done. But, through error or carelessness, the power end is often badly out of proportion, the tendency being to err on the side of furnishing too much power. The steam (or air) cylinder may be of such size as to

require a pressure of only 30 or 40 lbs. per sq. in., while the pipe-line pressure is 70 or 80 lbs., as usual with mine compressor plants. So it often happens that the deepest pump in the mine is the only one operating under a proper pressure. The cylinders of the others, even if running under throttle, are filled with air at full pressure when exhaust takes place.*

The difficulty with common direct-acting pumps is thus two-fold: the air is used without expansion, and the pressure is often higher than is necessary. Recognizing, however, the convenience with which the inexpensive, ready-made single-cylinder pumps may be installed, and that in many cases efficiency of operation is really a secondary consideration, a few points will here be discussed as to their employment, and the volume of air required for a given quantity of work. Questions relating to the expansive use of compressed air for pumps will be taken up afterward.

Cylinder Dimensions of Simple Pumps. In calculating the sizes of cylinders for a simple, or single-cylinder pump, to work under given conditions, the dimensions of the water cylinder must first be determined. There are three variables to be dealt with, *viz.*, diameter, length of stroke, and number of strokes per minute; or the last two factors named may be combined in the shape of piston speed per minute. The volume of water to be raised being given, the cylinder dimensions may be obtained from lists of standard sizes of pumps, which would usually be adhered to on the ground of saving in first cost. With a given air pressure and head of water, the diameter of the air cylinder obviously depends upon that of the water cylinder. The following relation between the two has been determined by Mr. William Cox:† "Area of air cylinder is to area of water cylinder as half the head is to the air pressure."

In using Table XLII, ratios for intermediate heads and pressures may be obtained by interpolation.

In this table the unit diameter of water cylinder is taken as 1 in. Diameters of air cylinders, as calculated, will be in deci-

* Some suggestive remarks on this subject are made by Frank Richards, "Compressed Air," pp. 171–172.

† *Compressed Air Magazine*, Feb., 1899, p. 583. (By permission.)

mals, and often of odd sizes not occurring in practice. After determining the exact diameter, the nearest standard diameter of cylinder would be chosen and the air pressure and piston speed adjusted accordingly.

TABLE XLII

RATIOS OF DIAMETER OF AIR CYLINDER TO DIAMETER OF WATER CYLINDER. (WILLIAM COX)

Head in Ft	Air Pressure, Lbs						
	20	25	30	35	40	45	50
50	1.12	1.00	0.91	0.84	0.79	0.74	0.71
100	1.58	1.41	1.29	1.20	1.12	1.05	1.00
125	1.77	1.58	1.45	1.34	1.25	1.18	1.12
150	1.94	1.73	1.58	1.45	1.37	1.29	1.22
175	2.09	1.87	1.70	1.58	1.48	1.39	1.32
200	2.24	2.00	1.82	1.69	1.58	1.49	1.41
225	2.37	2.12	1.94	1.79	1.68	1.58	1.50
250	2.50	2.24	2.05	1.90	1.77	1.67	1.58
275	2.62	2.35	2.14	1.98	1.85	1.75	1.66
300	2.74	2.45	2.24	2.07	1.94	1.82	1.73
325	2.85	2.55	2.33	2.16	2.02	1.90	1.80
350	2.96	2.64	2.42	2.24	2.09	1.97	1.87
375	3.06	2.74	2.50	2.31	2.16	2.04	1.94
400	3.16	2.83	2.58	2.39	2.23	2.11	2.00
425	3.26	2.92	2.66	2.46	2.30	2.17	2.06
450	3.35	3.00	2.74	2.53	2.37	2.24	2.12
475	3.44	3.08	2.82	2.60	2.44	2.30	2.18
500	3.53	3.16	2.89	2.67	2.50	2.36	2.24

Volume of Air for Pumps Working without Expansion. To determine the volume of free air for a single-cylinder pump, use the following formula:*

$$V = 0.093 W_2 \frac{h \times G}{P}, \text{ in which:}$$

V = volume of air in cubic feet per minute;
h = head in feet under which the pump is to work;

* *Compressed Air Magazine*, Feb., 1899, p. 581. (By permission.)

G = gallons of water to be raised per minute;
P = receiver gage pressure of air to be used;
W_2 = volume of free air corresponding to 1 cu.ft. at the given pressure, P.

In this formula, 15% has been added to the volume of air to cover losses. The following table gives values of W_2 and 0.093 W_2 for different pressures:

TABLE XLIII

Air Pressure, P, in Lbs.	W_2	0.093 W_2
15	2.02	0.18786
20	2.36	0.21948
25	2.70	0.25110
30	3.04	0.28272
35	3.38	0.31434
40	3.72	0.34596
45	4.06	0.37758
50	4.40	0.40920
55	4.74	0.44082
60	5.08	0.47244
65	5.42	0.50406
70	5.76	0.53568
75	6.10	0.56730
80	6.44	0.59890
85	6.78	0.63054
90	7.12	0.66216

For example, let it be required to find the volume of free air per minute required to raise 200 gals. of water to a height of 150 ft., the gage pressure being 30 lbs. From the table, 0.093 W_2, corresponding to 30 lbs. = 0.2827; hence,

$$V = 0.2827 \times \frac{200 \times 150}{30} = 282.7 \text{ cu.ft. free air.}$$

The horse-power may be calculated from Table XLIV, in which the mean pressures per stroke (from Table VII, Chap. X), for the different terminal pressures, are given in the second column, and the horse-powers in the third column:

TABLE XLIV

Terminal Pressure, Lbs.	Mean Pressure per Stroke	H. P. per Cu ft Free Air.
20	14.40	0.0628
25	17.01	0.0743
30	19.40	0.0847
35	21.60	0.0943
40	23.66	0.1033
45	25.59	0.1117
50	27.30	0.1196
55	29.11	0.1270
60	30.75	0.1340
65	32.32	0.1406
70	33.83	0.1468
75	35.27	0.1527
80	36.64	0.1583

As the horse-power corresponding to a given terminal pressure does not increase in constant ratio with the initial air pressure, it follows that the higher pressures are not so economical for simple pumps as low pressures. Expressed in another way, the work of compression decreases with the air pressure, and therefore the useful work done in a pump using air at full pressure is greater at low pressures and its efficiency is increased. Thus, in the example given above, the horse-power developed in using the 282.7 cu.ft. of free air, at a pressure of 30 lbs., is:

$$282.7 \times 0.0847 = 23.94 \text{ H.P.}$$

If the air pressure employed were 50 lbs., the cu.ft. of free air would be 245.52 and the corresponding H.P., 29.36, the added power cost being 5.42 H.P. It may be stated that the difference in favor of the lower air pressure is offset in part by the fact that, at the higher pressure, a pump with a smaller power cylinder will do the same work, thus saving in the first cost.

But the low pressures thus shown to be suitable for simple pumps would not serve for machine drills, which must be considered first, as they are in nearly all cases the chief users of compressed air in mines and quarries. To secure the best results from the pumps, a separate, low-pressure compressor would be

required, a provision which is usually out of the question. Since it is generally necessary to use high-pressure air, at, say, 80 or 90 lbs. gage, the air must either be wire-drawn into the pump cylinder or else reduced to the required pressure before being delivered to the pump.

In the first case, the results as to volumes of air used, as given in the preceding discussion and tables, must be modified by introducing a factor of increase, based on the ratio which the pressure to be used in the pump bears to the pressure carried in the air main. Edward A. Rix furnishes a table,* part of which is abstracted in Table XLV. It shows the volumes of free air theoretically required for a unit of 10,000 ft.-gals. of work (=83,000 ft.-lbs. or 2.5 H.P.), at different air pressures, referred to a standard receiver pressure of 90 lbs.

TABLE XLV

Gage Pressure, Lbs	Ratio of Compression, Referred to 90 Lbs	Cu ft of Air Calculated from Cox's Formula	Factor of Increase for Wire-Drawing from 90 Lbs	Increased Volume, Cu ft	Actual H P at 90 Lbs	Efficiency on Basis of 2.5 H.P. Theoretical.
20	3	113	1.26	142	28.6	9
25	2.6	108	1.22	125	25	10
30	2.3	97	1.19	115	23	11
35	2.1	93	1.17	108	21.5	11.6
40	1.9	89	1.14	102	20.5	12.2
45	1.7	87	1.12	97	19.7	12.7
50	1.6	85	1.11	93	19	13.1
55	1.5	82	1.09	89	18.2	13.7
60	1.4	80	1.07	86	17.4	14.3
65	1.31	79	1.06	84	16.8	14.9
70	1.24	78	1.05	82	16.4	15.3
75	1.17	77	1.04	80	16	15.6
80	1.1	76	1.03	78	15.6	16
85	1.05	75	1.02	76	15.2	16.4
90	1.0	74	1.0	74	14.8	16.9

The factors in column 4 are assumed as about 70% of the ratios of the absolute temperatures due to expansion of the air from 90 lbs., to the air pressures in column 1. They may be

* *Transactions Technical Society of the Pacific Coast*, Aug. 3, 1900.

taken to apply when the length of air main from the compressor to the pump is moderate, as in carrying the air to a pump situated at the bottom of an ordinary shaft. The showing is a poor one, but the unfavorable working conditions, as to the type of pump and mode of using the air, must be taken into account.

In the second case, the normal air pressure carried in the mine (say, 90 lbs.) may be reduced to a suitable pump pressure by placing a reducing valve in the air main. The increase of volume thus produced will be accompanied by a considerable drop in temperature, so that the full increase is not realized. Part of the lost heat will be regained by friction, and from external sources if there be any considerable length of pipe between the reducing valve and pump; but the efficiency will be materially increased if the cold, partly expanded air be passed first into an underground receiver and thence to the pump. This arrangement has been satisfactorily adopted, for example, in the case referred to at middle of p. 256. An adjustable spring-reducing valve is set to furnish any desired pressure below that in the main. That is, the volume of air allowed to pass is such as to maintain automatically a certain difference in pressure between that in the main and the pipe leading to the second receiver. The latter serves three purposes: (1) if it be of ample size or of the tubular type the air will regain nearly, if not quite, its normal temperature; (2) much of the entrained moisture will be deposited, and trouble from freezing avoided; and (3) the receiver, if placed near the pump, will minimize the pulsations and equalize the air pressure.

In the particular instance to which reference is here made, two underground receivers were installed, 300 ft. apart, the reducing valve being put in the main just above the first receiver. This arrangement not only caused a very complete deposition of the moisture, but the air entirely recovered its normal temperature by the time it left the second receiver on its way to the pump. The main air pressure was 85 lbs., and at the pump about 45 lbs. Indicator diagrams showed 128.5 H.P. developed by the compresssor and 16.45 H.P. at the pump, or an efficiency of 12.5%; thus agreeing quite closely with the figures in Table

XLV. Subsequently, by compounding one of the pumps, using 62 lbs. initial pressure in the high-pressure cylinder and admitting some live air to the intermediate pipe between the cylinders, the efficiency was raised to 25.9%. This must be considered a fairly satisfactory performance for a pump not specially designed for its work.

By adopting stage compression or by reheating, or both, the total efficiency can of course be increased considerably beyond the efficiencies shown in the table. Mr. Rix states that by actual test of a number of simple pumps he found their work to be approximately 135 ft.-gals. per cu.ft. of free air. For stage compression the efficiency is increased by 15% (giving, say, 155 ft.-gals.), and, by reheating, the 135 ft.-gals. is increased by the ratio of the absolute temperatures under which the pump works, without deducting the small cost of reheating.

Prevention of Freezing of Moisture. Though this subject has already been discussed at some length, several additional points may be noted in connection with pumping. Some benefit may be derived by leading a jet of water from the pump column into the air pipe, just before reaching the pump. A very small quantity of water will suffice to prevent an excessive drop in the temperature of the exhaust. A better way is to tap a ¼-in. pipe into the column pipe, draw down the end of this pipe to, say, one thirty-second of an inch and insert the nozzle so formed into the exhaust port. The author has observed the plan of carrying a small steam jet close to the exhaust port; but it is obvious that this is feasible only when steam is used near by for some other purpose. Moreover, steam so applied is utilized much less perfectly than when used in a cylinder jacket. If steam be available, a little may be injected into the feed air pipe near the pump. An intimate mixture between the steam and air is thus produced, and in condensing the latent heat of the steam is given up. If water at 212° F. be injected, each pound in cooling down to 32° F. will give up 180 thermal units. But with steam at the same initial temperature, each pound in condensing gives up 966 thermal units, in addition to the 180 units imparted in cooling to 32°. Still another mode of preventing

freezing is to warm the compressed air by passing it through a coil of pipe, placed in an enlarged section of the water column, or else in the pump suction pipe.

Compressed-Air-Driven Compound Pumps. It is a commonly held idea that if compressed air be used for operating compound, direct-acting pumps, it should be employed like steam, with a cutoff in each cylinder. The resulting drop in cylinder temperature would obviously be less than that caused in a single cylinder by the same ratio of expansion from a given initial pressure. But in aiming thus to attain a higher efficiency, by adopting the largest possible range of expansion, very low cylinder temperatures would still be produced. The loss of heat takes place chiefly within the cylinder, instead of in, and just outside of, the exhaust port, as is the case with pumps working at full pressure. Furthermore, though the same total fall of temperature occurs in either case, when the air expands within the cylinder the force of the exhaust is diminished by the low terminal pressure, and the ports are the more liable to be choked with ice.

In order to use the air expansively the necessity for reheating in some form is clearly indicated, aside from any question of gain in economy. Various plans have been tried of warming the cylinders by the application of external heat, such as developing them in hot-air jackets, surrounding them by water, even heating them by the flames of large lamps or torches. But air is too poor a conductor of heat to render these means efficient.

The mode of applying extraneous heat may be varied in several ways, *viz.*, (1) Preheating the compressed air sufficiently to permit of a reasonably early cutoff in each cylinder, while still avoiding too low an initial temperature in the low-pressure cylinder; (2) in addition to preheating, the air may be reheated between the cylinders; (3) using cold air at full pressure in the high-pressure cylinder and expanding into the low-pressure cylinder, with or without reheating; (4) using cold air at full pressure in both cylinders, the air being expanded between them, with the application of reheating.

The first two methods are feasible when the compound pump is of suitable design and the heating properly applied; but there

would be an undesirable variation in power and speed, for an engine necessarily working under a constant load, if the pump be of the usual direct-acting type, without fly-wheel. Moreover, under the first plan a high initial temperature would be necessary. If the expansion be adiabatic, from an initial pressure of, say, 80 lbs. to atmospheric pressure and normal temperature, the temperature to which the air would have to be preheated is given by the expression:

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \text{ or, } T' = 70^\circ + 459^\circ\left(\frac{80+15}{15}\right)^{0.29} = 446^\circ \text{ F.}$$

Although this temperature would be rapidly lowered during the stroke, proper lubrication of the cylinder might be interfered with. The third method would avoid in part the difficulty of variation in power and speed, though there would still be a variable back-pressure on the high-pressure piston; but the increase in volume due to clearance, and on expanding into the passages and intermediate pipe to the low-pressure cylinder, would considerably reduce the temperature of the air, and a large further drop would ensue during the work of expansion in the low-pressure cylinder. Such temperature drop may be prevented, or at least diminished, by introducing a receiver-reheater between the cylinders, with material gain in efficiency. This method has frequently been adopted, and on the whole is much preferable to the two first mentioned.

The fourth arrangement, however, appears to be the most satisfactory. As has been pointed out by E. A. Rix,* in the practical application of compressed air to pumps only a small part of the total possible work of expansion within the two cylinders can be realized, even in favorable circumstances. Nevertheless, if properly installed and operated, it becomes perfectly practicable to drive a compound pump by compressed air. It is a much more satisfactory machine than a single-cylinder pump, and is capable of working with a fair degree of efficiency. This may be accomplished by expanding the

* *Transactions Association of Engineering Societies*, 1900. Mr. Rix also proposes the use of three-, and even four-cylinder pumps.

air between the cylinders only, restoring the consequent loss of pressure by reheating and employing full pressure in both cylinders. Thus no drop of temperature takes place in the cylinders themselves, and the pressures, back-pressures, and speed are constant. Each air card is practically rectangular in shape. The pressure drop between the cylinders may be made small; in fact, it need not be more than is sufficient to give the head necessary to cause an active flow of air into the intermediate reheater and thence to the low-pressure cylinder. A drop of, say, 20 lbs. for an initial pressure of 70–80 lbs. will usually answer.

The degree of heat to be imparted by the intermediate reheater, to restore the heat lost by a drop of 20 lbs., would be only 204° F., for a final temperature of 60° at exhaust. If the pump be suitably situated, an ordinary fuel-burning reheater may be employed; or, should this be inadmissible, the water from the pump suction or column pipe may be utilized for reheating, as already suggested. An example of this arrangement, which has often been cited, is to be found in the Gwin Mine, Calaveras Co., California. A Worthington compound pump, having a capacity of 200 gals. per min., was installed on the 600-ft. level of the mine. Placed in the suction pipe of the pump is a 300-H.P. Wainwright heater, with corrugated copper tubes. The water in the pump, at a temperature of 60°–70° F., passes through the heater tubes on its way to the pump suction valves. The air, on being exhausted from the high-pressure cylinder, at a pressure of 35 lbs., passes into the heater and through the spaces between the tubes. In this way, the temperature of the air is raised practically to that of the water and, after expanding again in the low-pressure cylinder, is exhausted without freezing. Should the sump water be foul, the heater tubes must be cleaned from time to time; otherwise the coating of sediment materially reduces their conductivity. Still better results would be obtained from such an installation by employing a fly-wheel pump with a shorter cutoff. The lower temperature could then be met by water-jacketing both cylinders, the jackets being supplied with water by a small pipe

from the pump column. Though the quantity of heat thus restored to the expanded air is far smaller than that which would be derived from a fuel-burning reheater, this simple device is convenient and satisfactory for underground service.

By employing reheating in connection with properly designed and operated air-driven compound pumps, efficiencies of 40–50% may be realized. With 3-cylinder pumps, furnished with intermediate heaters, the efficiencies are still higher, reaching even 70%. Reference has already been made to the economic advantages of using the Cummings system of high-pressure transmission for operating compressed-air pumps.

CHAPTER XXV

PUMPING BY THE DIRECT ACTION OF COMPRESSED AIR

In the apparatus included under this heading, the compressed air acts directly on the water or other fluid to be pumped; there is no piston, cylinder, nor other moving part except the valves. The limiting depth or head at which these pumps will work depends on the gage pressure and mode of using the air. When operated under proper conditions and with expansive use of the compressed air, they compare favorably in first cost and efficiency with ordinary simple-cylinder piston pumps, and their maintenance cost is very low.*

There are three classes:

1. Pneumatic-displacement pumps, using compressed air with or without expansion.
2. The "Return-Air System."
3. "Air-lift" pumps, always working expansively.

Pneumatic-Displacement Pumps are of several kinds. In the type form, the compressed air acts directly upon the surface of the water contained in a submerged chamber or tank, valves being provided for controlling admission of air and water. The water is displaced by the air and is discharged from the tank through a pipe. There may be one or two tanks, the discharge pipe in the latter case being common to both. With one tank, the flow from the pipe is intermittent; with two, practically constant, the pair of tanks then having the same relation to each other as the chambers of the pulsometer pump. The working head is that which corresponds to the air pressure employed.

* A valuable text-book, "Pumping by Compressed Air," by E. M. Ivens, member of Amer. Soc. Mech. Engs., was published in 1914 by John Wiley & Sons. To this book the reader is referred for a fuller discussion of the theory and practice of the subject than is appropriate here.

Besides being useful for all low-head service, as pumping from wells or into tanks, pneumatic-displacement pumps, having practically no moving parts, are distinctly advantageous for pumping acids, chemical solutions, etc., which would rapidly destroy a piston pump.

The double-chamber Merrill pneumatic pump is shown diagrammatically by Fig. 218. Compressed air enters through an

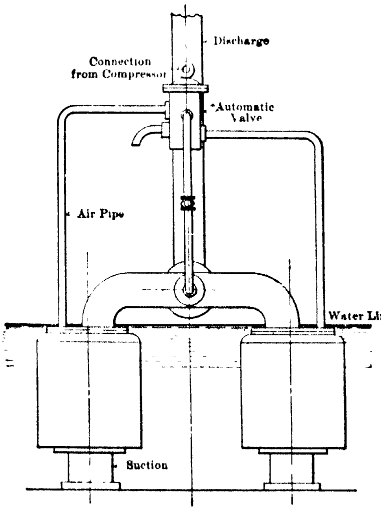


FIG. 218.—Merrill Pneumatic-Displacement Pump

automatic valve, which opens connection alternately with the water chambers. The air pressure required depends on the height of lift. Since the pressure per sq. in. of a column of water is 0.434 lb. per ft. of head, the height to which a given air pressure will raise water is equal to the gage pressure divided by 0.434; thus, air at 80 lbs. will pump to a height of $\frac{80}{.434} = 184$ ft. To cover friction, leakage, absorption of air by the water, and to provide dynamic head for overcoming inertia and securing a proper speed of discharge, an added air pressure

is necessary. In terms of volume, 1 cu.ft. of water will be displaced per cu.ft. of compressed air. One cu.ft. of air at 80 lbs. $= \frac{1 \times (80+15)}{15} = 6.33$ cu.ft. free air. To this should be added for losses, etc., say 20%, making a total of 7.6 cu.ft. free air per cu.ft. of water. Taking 1 gal. of water equal to 0.134 cu.ft., the work done per cu.ft. of compressed air, against a head of 184 ft., will be: $\frac{184}{0.134 \times 7.6} = 180$ ft.-gals. $= 1,503$ ft.-lbs. In some cases more than 20% should be allowed. The actual work done in compressing 1 cu.ft. of air to 80 lbs. gage, by a single-stage compressor (Table V, Chap. X) is 0.183 H.P., or 6,039 ft.-lbs.; hence, the efficiency of the pump, with the above allowance for losses, is nearly 25%, which compares favorably with the efficiencies of simple direct-acting pumps.

The displacement pump in its usual form exhausts at each stroke a tankful of air practically at gage pressure. By employing a series of these pumps in a shaft, and using the air expansively, the possible height of lift with a given initial pressure and the total efficiency, will greatly exceed that shown above.* This can be done by a suitable valve control, by which the air is expanded from the lowermost tank to the one next above, and so on, for smaller and smaller lifts toward the top of the series. When the last tank is discharged, the whole system is occupied by expanded air, at a pressure of 2 or 3 lbs., which is then exhausted into the atmosphere. Air is admitted by the valve at intervals into the lowest tank, and the working of the system proceeds automatically. At 80 lbs. air pressure, water can thus be raised to a height of about 330 ft., instead of 184 ft., as in the preceding example, and at an efficiency of about 40%.

Another displacement pump is the Latta-Martin (Fig. 219), designed chiefly for raising large volumes of water under low heads, though it may be constructed for any desired air pressure and head.† A pair of submerged tanks take water through

* This series system of tanks has been proposed by E. A. Rix, *Trans. Tech. Soc. of the Pacific Coast*, Aug. 3, 1900, p. 187.

† See also *Compressed Air Magazine*, Jan. 1907, p. 4342.

large disk valves in the bottom. The valve mechanism, admitting air alternately into each tank, comprises a main and auxiliary valve, each thrown by a piston valve, like those of single-cylinder steam pumps. The valves are thrown by the oscillations of a pair of levers, from each of which is suspended a bucket filled with water and hanging in a housing within the main tank. When the pump is in operation, the bucket housings are alter-

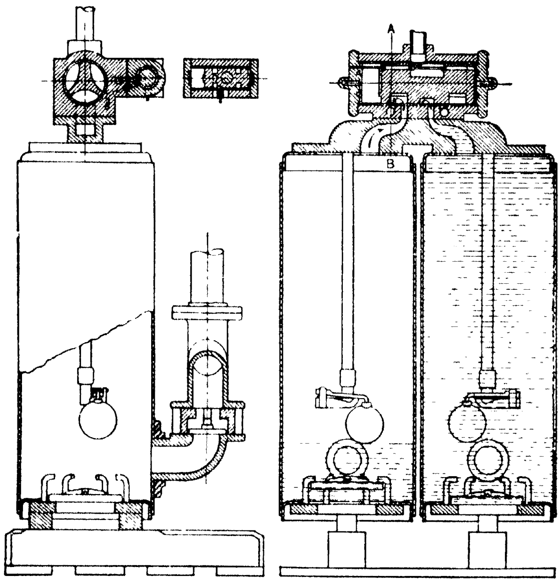


FIG. 219.—Latta-Martin Displacement Pump.

nately filled and emptied, so that the difference in effective weight of the buckets causes them to rise and fall.

The Halsey pneumatic pump (Pneumatic Engineering Co.) has a single, submerged tank, with a simple, automatic valve-motion, operated by a float.

If a displacement pump be required to work in acid water, as in mines containing sulphide ore, the tanks may be lined

with concrete and the other parts made of bronze; or the tanks may be replaced by excavations in the rock, adjacent to the shaft and lined with concrete or asphalt.

Return-Air Displacement System (Ingersoll-Rand Co.) uses compressed air expansively. Two tanks, either submerged or within suction distance of the sump, are connected by pipes with the compressor. If not submerged the water enters by siphon action. Compressed air admitted to one tank forces out the water (or other fluid) through a check valve to the pipe-line. Meanwhile, the compressor draws air from the other tank. The charge of air is so regulated that, when one tank is empty, the other is full; then a switch valve reverses the action of the tanks.

Referring to the diagram (Fig. 220), A and B are the tanks, with their air pipe-lines A^1, B^1; C^1 and C^2 are discharge check valves, preventing return of the ejected fluid; D^1, D^2 are check valves, preventing discharge through the inlets; E is the discharge pipe of both tanks; F is the automatic switch controlling the pumping cycle; J is an automatic compensating valve, which keeps the system supplied with air, taking care of loss and leakage. Instead of being exhausted into the atmosphere at each stroke, the compressed air after doing its work is conducted back to the compressor intake and expands behind its piston. Hence, the system is a closed one, the same air being used over and over, as in the Cummings return-air compressor plant (Chap. XVII).

In starting, after the water in one of the tanks has been expelled, the switch reverses and connects this tank with the compressor intake. Then, while the second tank is being discharged, the air exhausted from the first returns to the compressor and, acting expansively upon the intake side of the piston, reduces by so much the power required to drive the compressor. When the pressure in the first tank has fallen sufficiently (by being in communication with the compressor intake), it will again fill with water. Thus, the compressor transfers the same body of air from one tank to the other. Valve J is set to open during the suction period, at a negative pressure

a little greater than that required to draw water into the tanks. The switch-valve is operated automatically, by a device acting at the intervals required to complete a cycle in both tanks, or by an electric make-and-break mechanism, controlled by a pressure

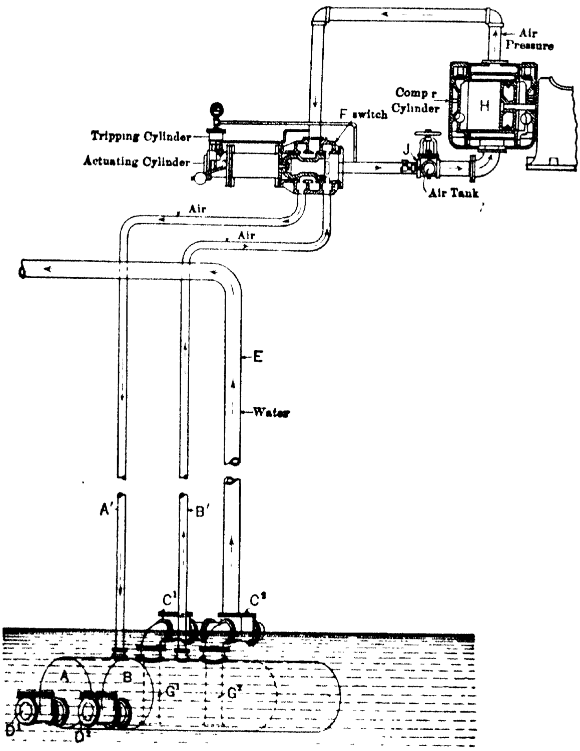


FIG. 220.—" Return-Air " Displacement System (Ingersoll-Rand Co.)

gage on the air intake. In the first case, a piston valve is operated by a small air cylinder, compressed air being admitted alternately to each side of the piston in the latter through an auxiliary valve. The volume of air for a given size of tank may be determined in terms of revolutions of the compressor.

This pumping system has an efficiency of 55–60%, and requires little attention during operation. For further details, see paper by Prof. Elmo G. Harris, *Trans.* Amer. Soc. C. E., Vol. 54.

TABLE XLVI

RETURN-AIR SYSTEM. SIZE OF COMPRESSOR, PIPING, ETC., FOR DIFFERENT HEADS, BASED ON 100 GAL. WATER PER MIN. SIZES FOR OTHER QUANTITIES OF WATER ARE PROPORTIONAL

Lift, Ft.	Capacity of Compressor, Cu ft per Min Piston Displacement for 100 Gals per Min.	Max I H P of Air Cylinder	Max I H P of Steam Cylinder	Aver H P of Steam Cylinders	Area of Air Pipe, Sq in. for Each 100 Gals Capacity	Area of Water Pipe, Sq In for Each 100 Gals Capacity.
50	39 84	2 74	3 22	2 80	96	7 70
60	42 78	3 28	3 85	3 37	1 03	8 25
70	45 30	3 85	4 53	3 93	1 09	8 73
80	47 70	4 45	5 22	4 49	1 14	9 12
90	49 80	5 03	5 91	5 05	1 20	9 60
100	51 84	5 07	6 07	5 61	1 25	10 00
120	55 44	6 96	8 18	6 73	1 33	10 60
150	59 94	9 00	10 60	8 41	1 44	11 50
170	62 64	10 42	12 25	9 54	1 50	12 00
200	66 12	12 76	15 00	11 22	1 58	12 65
250	71 10	16 46	19 35	14 02	1 71	13 70
300	75 06	20 45	24 00	16 83	1 80	14 40

This table assumes that the tanks are fully submerged.

Air-Lift Pump. This is a revival of an old principle. Since 1888, when Dr. Julius Pohlé proposed its application for pumping, the air-lift has become increasingly important. Besides its use for raising water from deep wells, it is applicable to a limited extent to pumping in shafts, and is well adapted for elevating finely divided pulpy material mixed with water, as the slimes and sands of cyanide and concentration mills.

The pump consists essentially of two pipes; a large column or delivery pipe and a relatively small air pipe, leading from the compressor receiver (Fig. 221). The delivery pipe, open at both

ends, is submerged to a depth proportionate to, but always greater than, the height to which the water is to be raised. The compressed-air pipe passes down to a point near the bottom, and admits air to the lower end or foot-piece of the delivery pipe. (Modifications of this arrangement are noted hereafter.)

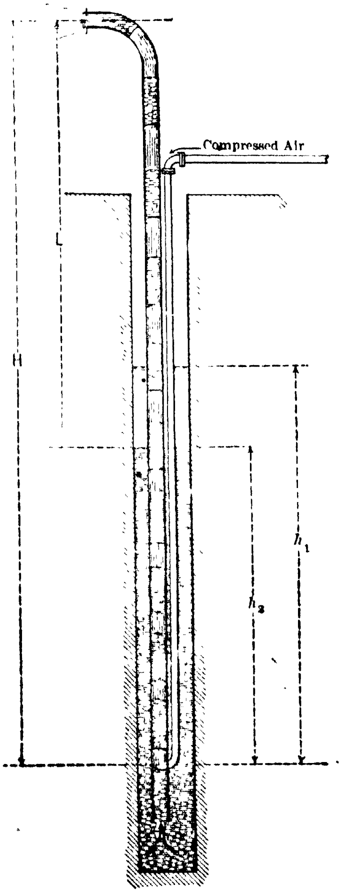


FIG. 221.—Diagram of Pohlé Air-Lift Pump.

In some respects the operation of the air-lift pump is the reverse in principle of the method of compressing air by the direct action of falling water (Chap. XV). If the discharge pipe be of very small diameter, the compressed air entering the bottom tends to form piston-like layers, which rise rapidly, alternating with masses of water (as is shown by experimenting with glass tubes). If the discharge pipe is of large diameter, the air is best admitted through a series of ports or nozzles, resulting in a more complete dissemination of the air through the mass of rising water. The water is raised chiefly by the aeration of the column of water, which causes a reduction in its specific gravity; added to this is the expansive force and vis viva of the

compressed air. Before the air is turned on, the water stands at the same level inside and outside of the delivery pipe. On entering the foot-piece, the air is under a pressure due to the weight of the rising column of water. As the bubbles of air rise, they expand with the decrease in head, and, on reaching the point of discharge, the tension of the air is reduced practically to atmospheric pressure. The initial air pressure required depends on the pressure due to head, measured from the point at which the air enters the delivery pipe to the surface of the water. At too high a pressure loss of work ensues at the compressor. Should the delivery pipe be too deeply submerged, in proportion to the net height of lift, an uneconomically high pressure will be required to force the air into the foot-piece; and, with insufficient submergence, more air will be necessary to produce the velocity of delivery.

Referring to Fig. 221, let:

h = depth of the delivery-pipe foot-piece below the normal water-level, before pumping begins, or when the water is at rest;

h_2 = height of water-level when the pump is in operation;

H = height of the column of mixed air and water, measured from the air inlet to point of discharge;

L = net height of lift = $H - h_2$.

The compressed air enters the foot-piece at a pressure P′, corresponding to the head h_2; or, $h_2 \times$ pressure per foot of hydraulic head = $0.434\ h_2$. Assuming that the water rises in piston-like masses (as would be the case with a single air nozzle and a delivery pipe of small diameter), the sum of the length of these masses in the column H must theoretically be equal to the outside solid column of water h_2 (the weight of the compressed air contained in the column being neglected). But, to overcome the frictional resistance and produce flow, the head h_2 must be greater. Under ordinary working conditions, the net height of lift L is found to be from $0.5\ h_2$ to say $0.65\ h_2$. Taking the second value and transposing: $h_2 = \frac{L}{0.65}$; and by substituting in the

expression for the value of P', as above: $P' = 0.434\frac{L}{0.65} = 0.67$ L. If, for example, L be 50 ft., $P' = 33.5$ lbs., and $h_2 = \frac{50}{0.65} = 77$ ft.

Since the air in the column H is divided into small masses, surrounded by water, its expansion during the upward flow may be assumed to be isothermal. If P' be its initial pressure, the mean pressure for the entire lift $= P \times \text{Nap.log}\left(\frac{P'}{P}\right)$, P and P' being absolute pressures. In the above example, taking P as 15 lbs., $P' = 33.5 + 15 = 48.5$ lbs., whence, the mean pressure $=$ 17.5 lbs. gage.

For starting the pump, the air pressure must be sufficient to overcome the normal static head h_1, but, when the flow has begun, the pressure required falls to that corresponding to h_2. Though this difference in pressure $(h_1 - h_2)$ may be considerable, it is readily met by temporarily speeding up the compressor. To minimize fluctuations between h_1 and h_2, the top of the well or sump should be extended laterally, in order to furnish a large horizontal area of water, the level of which would be but little affected by stoppages or by variations in air pressure and delivery. The throttle valve in the air pipe may be regulated by a float on the surface of the water. Care should be taken in the design of the foot-piece and in properly proportioning the air pressure to the submergence and net lift. Otherwise, air may leak back into the sump or outside column of water; and, if this becomes aërated, much more power and a larger volume of air will be required to keep the pump in operation, thus reducing the efficiency.

Since 1889 many experiments by competent engineers have been made on the air-lift pump. Among the first were those of B. M. Randall and H. C. Behr, on a 60-ft. well, with a stage compressor. A summary of these tests is given by E. A. Rix, in the *Transactions of the Technical Society of the Pacific Coast*, Aug. 3d, 1900, p. 206. In 1894 a series of tests were made at De Kalb, Ill.,* and in 1893 and 1896 on four pumps at Rockford, Ill.†

* *Eng. News*, July 12, 1894. † *Eng. News*, March 4, 1897.

The last-named were carefully carried out and the results compared in tabulated form. The heights of lift above water-level were 66.5, 90, and 91.5 ft., the air pressure being 76 lbs. gage and the submersion 225 ft. Both air pressure and depth of submersion appear to have been unnecessarily great. With a compressor of 124 H.P., the net work done was 24 H.P., or an efficiency of about 20%. With 600 cu.ft. free air per min., 200 cu.ft. of water were pumped, or 3 air to 1 water. The sizes of piping used were: delivery pipes, 4 ins., 5 ins., and 6¼ ins., with air pipes from 1½–2½ ins. In several of these tests the air pipe terminated in a ⅝-in. nozzle. The plan was also tried of closing the lower end of the air pipe and discharging the air through slot-shaped perforations in the sides near the bottom; but the results were inferior to those obtained from the single-nozzle opening. Possibly better work would have been done by some different arrangement or size of slots; for large pipes and volumes of water the single nozzle has not been satisfactory.

E. E. Johnson gives a table for computing the performance of the air-lift pump, including consumption of power and theoretical and total efficiencies for different heights of lift,* from which Table XLVII is abstracted.

These figures, which represent the work of well-proportioned plants, as to depth of submergence and air pressure, show that the efficiency falls off rapidly as the air pressure and height of lift increase. Under normal conditions and with small lifts, efficiencies of 30 to 35% are readily obtainable, and may rise to 45 or 50% with proper air pressures and ratios of submergence to height of lift.

Volume of Air for Air-Lifts may be computed by the following formula, which closely approximates average practice.†

$$V_a = 0.8\left(\frac{h}{C \times \log \dfrac{H+34}{34}}\right)$$

in which: V_a = vol. free air for raising 1 gal. water; h = total lift, ft.; H = submergence when running, ft.; C = constant.

* *Eng. News*, April 22, 1897.

† Ingersoll-Rand Co., suggested in part by E. A. Rix.

Values of " C " with proper submergence:

Lift in Ft. (h)	Constant
10– 60 ft.	245
61–200 "	233
201–500 "	216
501–650 "	185
651–750 "	156

TABLE XLVII

Lift		Theoretical Horse-power				Efficiency of Air-lift.				
		To Lift 1 Cu ft of Water per Minute	To Deliver 1 Cu ft of Air per Minute			Theoretical.			Total Efficiency from Power Applied to Water Del'd.	
Lbs. Pressure.	Ft Head.		Isothermal	Two-Stage.	Adiabatic.	Isothermal Compression	Two-Stage Compression.	Single-Stage or Adiabatic Compression.	Two-Stage Compression.	Single-Stage or Adiabatic Compression.
5	11 54	02185	.02514	02572	0263	87	848	83	623	.497
10	23 09	.04363	05586	05992	064	78	728	684	546	41
15	34 63	06545	09105	0962	1015	72	687	648	515	389
20	46 20	08727	12994	1391	1483	675	627	59	47	.354
25	57 75	109	17191	1897	2004	635	575	545	.432	327
30	69 31	13091	21678	2370	2573	603	548	508	.412	305
35	80 86	1527	26445	2915	3187	577	52	478	.39	287
40	92 41	17454	31375	3489	3842	557	502	455	.376	273
45	103 90	1963	36368	4085	4535	540	482	433	362	260
50	115 50	21818	41848	4722	5201	522	404	415	.348	249
55	127 00	24	47112	5366	6023	51	447	40	.336	24
60	138 60	26181	52855	6051	6818	495	432	384	324	231
65	150 10	2836	58612	6734	7008	483	422	372	316	223
70	161 70	30545	64812	748	8483	471	408	36	.307	216
75	173 30	3273	70952	.823	9380	462	398	.35	.299	210
80	184 80	3491	76843	898	1 0291	455	39	343	.292	206
85	196 30	37	83039	.976	1 1231	446	38	331	.285	198
90	207 90	3927	89444	1 055	1 2176	439	373	324	.28	.194
95	219.40	4145	96164	1 137	1 3148	431	268	315	.276	.189
100	230.90	43630	1 0243	1 247	1 4171	428	352	.308	.264	.185
110	254.10	.48	1 162	1 394	1.626	.413	346	.296	.26	.177
120	277.20	.5236	1 301	1 571	1 841	402	333	285	.25	.171
130	300.40	5675	1 443	1 755	2 068	394	324	.275	.243	.165

Air Pressure. Starting pressure is equal to the depth of foot-piece in the well, less the water head in ft. × 0.434. Working pressure is equal to the depth of foot-piece, less the pumping head × 0.434 + friction in air pipe + 2 lbs. back pressure in the nozzle or foot-piece.

Ratio of Lift to Submergence. The following figures represent average practice.*

For lifts up to	50 ft.,	70–66%	submergence.
"	50–100 "	66–55%	"
"	100–200 "	55–50%	"
"	200–300 "	50–43%	"
"	300–400 "	43–40%	"
"	400–500 "	40–33%	"

Tests. In several tests at Wandsworth, England, on a modified Pohlé air-lift, with a delivery pipe of increasing diameter toward the top, the total height of the delivery pipe was 580 ft., of which 324 ft. were submerged, the net lift thus being 256 ft. In this case, the distance $h_1 - h_2$ (Fig. 221) was 69 ft., air pressure, 135 lbs., ratio of volume of free air used to water discharged, 5.8 and 5.6 :1; total efficiency, 36%. This indicates an advantage in using a tapering column pipe.

The following results of a test on a 300-ft. well show the low efficiency of high lifts:†

Elevation of discharge above mouth of well	85 ft.
Depth to water-level during operation of pump	44 ft.
Net lift, water-level to point of discharge	129 ft.
Submergence of delivery pipe	248 ft.
Air admitted to delivery pipe 5 ft. above inlet end.	
Diameter of delivery pipe	3.5 ins.
Diameter of air pipe	1.25 ins.
Volume of water delivered per minute	82.5 gals.
Volume of free air used per minute	81.8 cu ft.
Gage pressure of air	107 lbs.
Consumption of free air per cu.ft of water	7.44 cu.ft.
Horse-power consumed by compressor	12.1
Total efficiency	22.3%

* Recommended by Sullivan Machinery Co

† G. C. H. Friedrich, *Trans. Ohio Soc. of Mech., Elec., and Steam Engrs.*, 1906. For data on a number of other air-lifts, see Peele's "Mining Engs. Handbook," 1918, Section 15, Art. 23.

Though the question of relative sizes of air and delivery pipes has not yet been satisfactorily answered, it is probable that ratios of diameter from 1 : 2 up to 1 : 2¼ or 3 will be found suitable. The absolute diameters are determined on the basis of frictional loss caused by the flow in the pipes. A water velocity of 250–300 ft. per min. may be assigned for the delivery pipe. For the friction losses in air pipes, see Chap. XVI. When the water is delivered at a distance from the pump, the added frictional resistance must be determined, and the air pressure and submergence correspondingly increased (see a paper by H. T. Abrams, in *Compressed Air Magazine*, Aug., 1906, p. 4135).

Table XLVIII gives some calculated values for air-lifts.

TABLE XLVIII

Lift, Ft.	Volume of Air per Cu ft. of Water.	Submergence, at 60% of Total Height of Delivery Pipe.	Air Pressure	H P per Gal Water per Min.
25	2	38	17	0 0184
50	3	75	33	0 0426
75	4 5	113	49	0 0828
100	6	150	65	0 1320
125	7 5	188	82	0 1910
150	9	225	98	0 2544
175	10 5	263	115	0 3150
200	12	300	130	0 3808

Foot-Pieces for Air-Lifts are varied in design (Fig. 222). At *A* is shown the original Pohlé side inlet, now rarely used. The annular foot-piece *B* (Ingersoll-Rand Co.) is a modification of *A*, admitting air all around the periphery of the delivery pipe, without contracting the discharge area. It is furnished for capacities of 20–325 gals. per min. In the Saunders foot-piece *C*, for a cased well, the air passes down through the annular space between the casing and delivery pipes. Another form of the Saunders foot-piece admits air to a hollow base plate studded with a group of vertical ¼-in. pipes, 18 ins. long. The mass of water is thus split up and the air mixed with it at the point of entering the delivery pipe. Design *D* is a variation of *C*.

The design shown at *E* (Fig. 222), made by the Sullivan Machinery Co., and both of those in Fig. 223 (Ingersoll-Rand

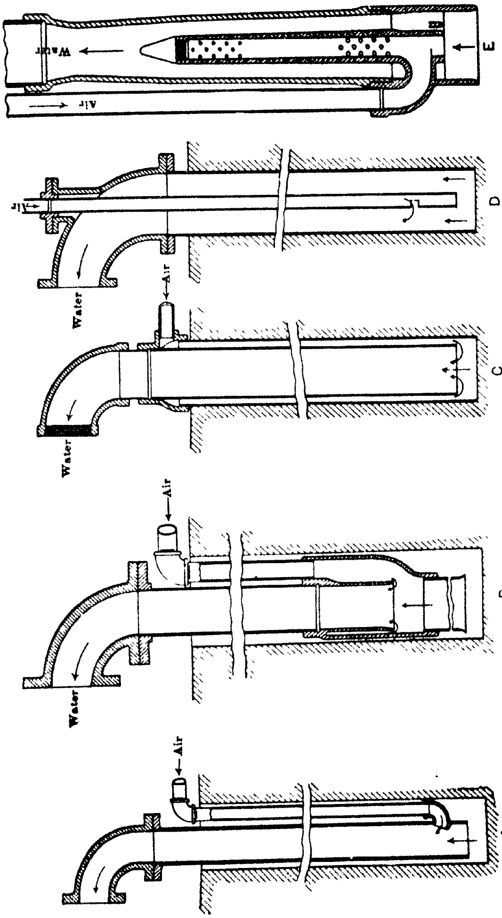


FIG. 222.—Foot-Pieces and Modes of Piping for Air-Lift Pumps.

Co.), admit air to the delivery pipe through numerous small holes, to secure a thorough intermixture of the air and water. The Sullivan foot-piece is a bronze casting, and is furnished for

capacities of 5–1,500 gals per min. In those shown in Fig. 223 the outer casing is steel, the inner tube of brass; their capacities are 15–2,000 gals. per min. To give an accelerated velocity to the mixed air and water on entering the delivery pipe, the three designs last mentioned have a Venturi contraction throat above the foot-piece.*

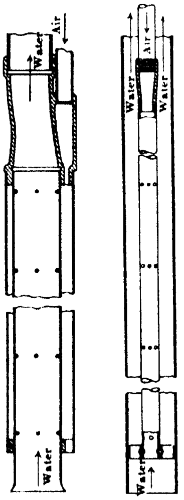


FIG. 223.—" VA " and " VC " Foot-pieces (Ingersoll Rand Co.)

Elaborate tests on the air-lift pump were made in 1907 by Messrs. Henderson and Wilson at the two 200-stamp mills of the Angelo and Cason mines, of the East Rand Proprietary Mines, Limited, South Africa.† At these mills slimes and sands are raised to the settling tanks by air-lifts instead of the usual tailings-pumps and wheels. The delivery pipes used in the 19 recorded tests were of two kinds, *viz.*, 10- to 16-in. pipes of constant diameter, and several pipes increasing in diameter from 12 and 14 ins. at the bottom to 14 and 16 ins. at the top. As a uniform taper was impracticable, the latter pipes, for a length of 35 ft. above the air inlet, were lined with 1 in. of wood, which served also to protect the metal from the scouring action of the mixed sands or slimes and water.

The foot-piece used in the earlier tests was flared out and closed at the bottom, the water and pulp being admitted through 4 large ports, 2½ ft. below the air inlet and having a combined area of about 200 sq. ins. The air inlet was a single opening, 4 ins. diameter. For the later tests, the foot-piece was open at the bottom and flared out to double the diameter of the column

* For further details of air-lift apparatus see the catalogues of the Sullivan Machinery Co., and Ingersoll-Rand Co., which contain much useful information.

† For full details see *The Engineer* (London), Jan. 10, 1908, p. 26.

pipe, to increase gradually the velocity of inflow (Fig. 224). A ring of 12 holes, 1 in. square, admitted the air.

The modified foot-piece was supported on timbers, so that the entire bottom was open for the free admission of the material to be pumped. The column pipe was of steel tubing, expanded

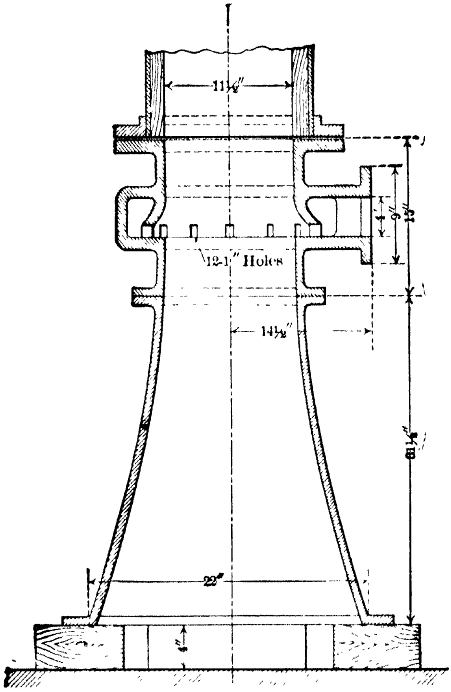


FIG. 224.—Foot-Piece for Air-Lift Pump, for Raising Mill Tailings and Slimes.

into cast-iron flanges, and lined in the lower part with wood. This design gave materially higher efficiencies than the one first used. Table XLIX, though presenting the details of only 4 of the 19 tests made, indicates the general results obtained. These details show that the air-lift, when properly designed for stated conditions, can compete successfully with the tailings

wheel, in common use in the district, and that it is superior to the tailings pump.

TABLE XLIX

	Test	1	2	3	4
Conditions.	Number and size of delivery pipes.	Two 10-in	Two 10-in.	One 16-in, decreasing to 14 ins.	One 14-in, decreasing to 12 ins
	Submersion in ft	32.75	35.75	37.75	48.85
	Lift in ft	32.5	29.5	27.5	27.09
	Ratio of submersion to lift	1.000-1	1.21-1	1.372-1	1.77-1
	Gage pressure of air, lbs	15	16	17	22
	Kind of foot piece	Original	Original	Modified	Modified
	Throat diameter of foot piece.	10 ins	10 ins	13½ ins	11½ ins.
Performance.	Free air, cu ft per min	2250	1270	740.48	846
	Free air, per cu ft. of slimes	7.27	4.06	2.74	2.64
	Cu ft of slimes per min	310	315	200	320
	Throat velocity, cu ft per sec	4.7	4.8	4.85	7.39
	Theoretical H P. in pulp raised	10.3	17.8	15.23	16.6
	H P. per cu ft free air compressed	.048	.050	.053	.064
	Air horse power	[illegible]	[illegible]	42.21	54.14
	Efficiency, per cent	[illegible]	[illegible]	37.15	30.55

The conditions were modified in the successive tests, as to the ratio of submersion to lift, diameter of delivery pipe, and air pressure. As a basis for computing the horse-power represented by the mixture of water and pulp raised, the weight of the slimes was determined to be 63.3 lbs., and of the sands, 64.56 lbs., per cu.ft. Thus, for the sands, the horse-power was:

$$\frac{(\text{Quantity of sands}+\text{water})\times 64.56\times\text{ft. lift}}{33{,}000}=.001956\times Q\times\text{ft. lift.}$$

The term "sands" refers to the mixture of water and ore as crushed by the stamps, from which the "slimes" have been separated in the milling process.*

* For more data on air-lifts for mill pulp, see *Compressed Air Magazine*, May, 1914.

Lansell's Air-Lift is a modification of the air-lift, applied by Mr. George Lansell to pumping from a deep mine shaft in the Bendigo district, Victoria, Australia. A series of lifts were used for a depth of 1,385 ft. Fig. 225 shows diagrammatically two of the lifts.

The compressed air is conveyed from the receiver in a pipe *A*, running down the shaft. The water is conducted from a tank through a pipe *D*, which first passes down the shaft a certain distance, depending upon the height to which the water is to be raised, and is then connected with an enlarged section of pipe *E*, at the foot of the delivery pipe *B*. Thus, the piping for each lift forms an inverted siphon. At the lowest point of the siphon a short branch pipe *C* enters from the air main *A*, the end of this branch being directed upward into the foot-piece *E*. Before the compressed air is turned on the water stands at the same level in pipes *D* and *B*. The effect of this arrangement is like that produced by submerging the lower part of the delivery pipe, as in the ordinary air-lift. Check valves are placed, as shown, in pipes *D* and *C*, to prevent air or water from passing back into the air pipe or into the tank. A throttle valve in pipe *C* regulates the supply of air. The relative heights of the various parts are variable, the dimensions shown on the sketch indicating substantially the proper depth of the inverted siphon below the tanks, and the corresponding height of lift; thus, from the tank at the 250-ft. level, the pipe *D* passes down the shaft 140 ft., to the foot of the delivery pipe which discharges at the surface. By a series of lifts the water may thus be raised from any desired depth. The air pressure is the same for all, this pressure being 60–80 lbs. per sq. in., or that which is ordinarily furnished for mine service.

Air-Lifts at the Old Dominion Copper Mine, Arizona.* Ten inch air-lifts (Fig. 226) were installed from the 12th to the 10th level (200 ft.) and from the 10th to the drainage tunnel (431 ft.). The air lines were 4-in. The submergence on the lift from the 12th level was 177 ft., or 47%; capacity, 1,650 gals. per min.; air pressure, 80–90 lbs.; air consumption, 1,080 cu.ft. per 1,000

* P. G. Beckett, *Trans.* A.I.M.E., Vol. 55 (1916), p. 53.

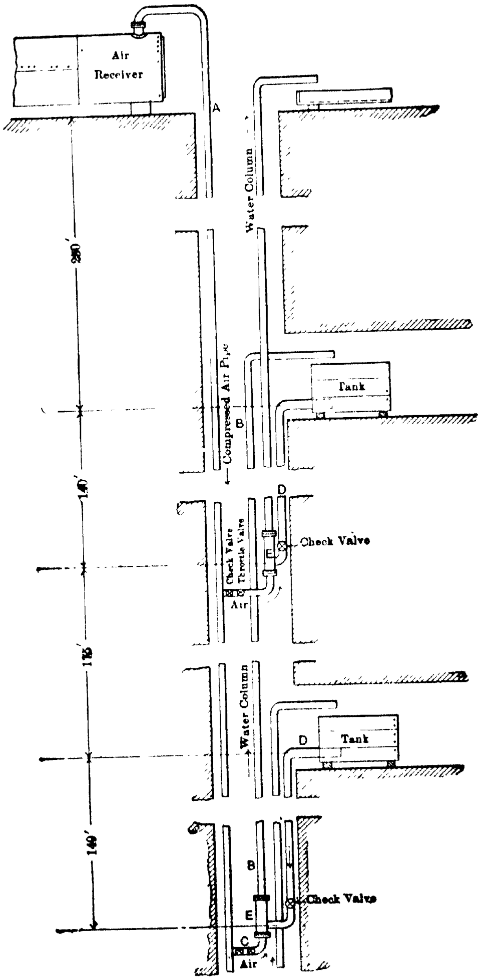


FIG. 225.—Diagram of Lansell's Air-Lift Pump for Mine Shafts.

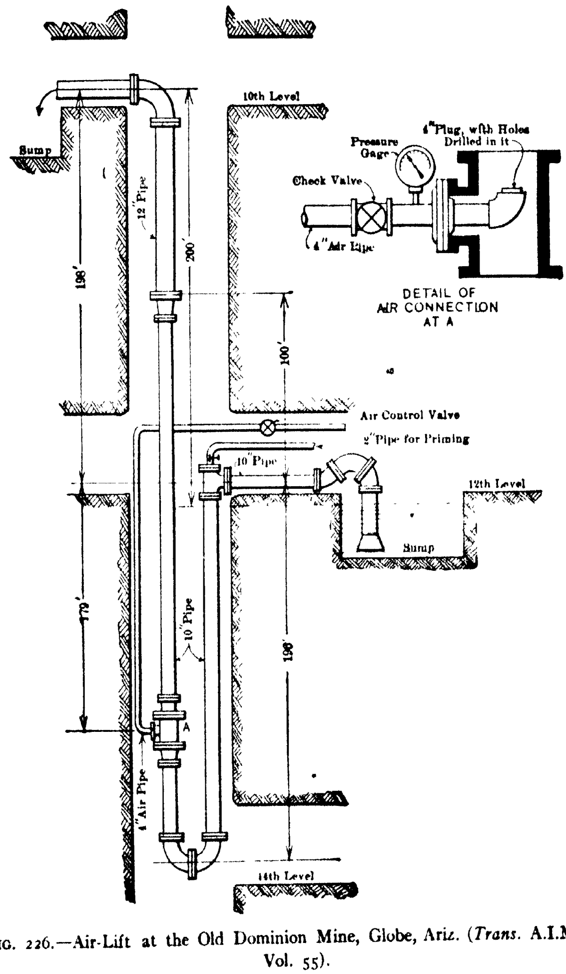


FIG. 226.—Air-Lift at the Old Dominion Mine, Globe, Ariz. (*Trans.* A.I.M.E., Vol. 55).

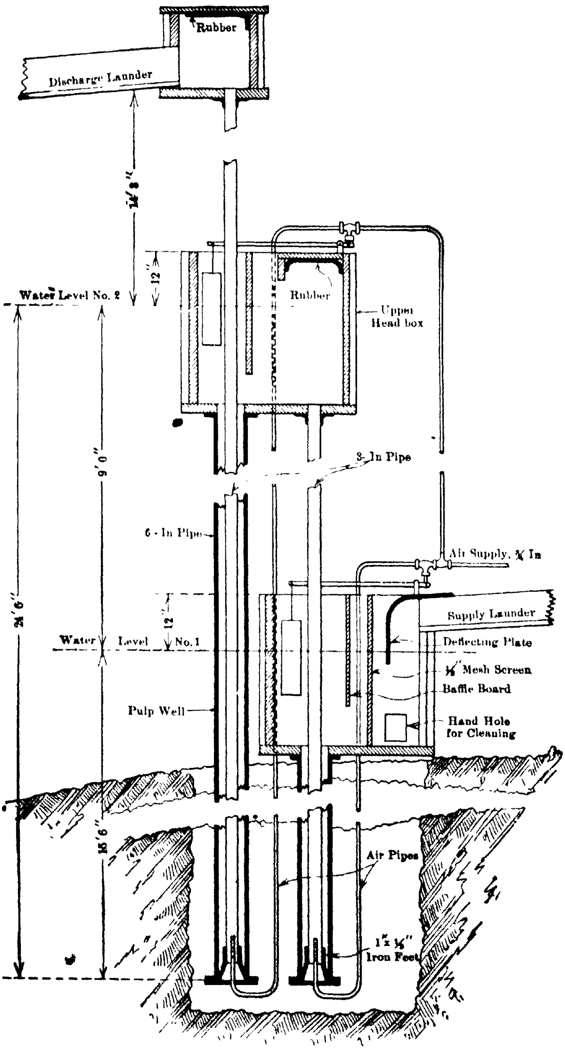


FIG. 227.—Two-Stage Air-Lift, Coolgardie, Western Australia.

gals.; maximum efficiency, 36%. Submergence on the 10th level lift was 188 ft., or 30.4%; capacity, 1,233 gals. per min.; air pressure, 90–100 lbs.; air consumption, 2,681 cu.ft. per 1000 gals.; maximum efficiency, about 30%.

During a period of unwatering the 16th level of the mine, the columns of two direct-acting pumps were converted into air-lifts.

Two-Stage Air-Lift at Burbanks Main-Lode mine, near Coolgardie, Western Australia, for raising and transporting mill sands (Fig. 227).* The height of lift for single-stage would have required sinking a pit 39½ ft. deep, for the necessary submergence; for two-stage, the pit was only 15½ ft. Above the pit are two wooden head-boxes, with water levels 9 ft. apart. The wells for the pulp are of 6-in. cast-iron pipe; inside pipes are 3 in., and the air pipes ¾ in. diameter. The air outlets of the latter are slots, at an angle of 45°, thus giving the air jets an upward direction. Supply of air is regulated by floats, so that a nearly constant level is maintained in each head-box, and no air is blown to waste. As the sand is heavy, the pulp mixture is 2½ water to 1 sand. Results obtained are as follows: free air per min., 47 cu. ft.; sand per min., 320 lbs.; water per min., 800 lbs.; sand per cu.ft. free air, 6.8 lbs.; sand and water per cu.ft. free air, 23.83 lbs.

NOTE.—In 1910 and 1920, several large air-lifts were designed and installed by the Supt., S. F. Shaw, for unwatering the *Tiro General Mine*, Charcas, Mexico.† The net lifts have ranged from 160 to 1080 ft., and the water pumped from 275 to 900 gals. per minute, pumping efficiency, 20% to nearly 40%.

As the work is still in progress (April, 1920), discussion of the operating results must be postponed, but it should be stated that these results show that the large percentages of submergence customary hitherto are not in reality necessary. At 1,060 ft. lift, and submergence of only 30 ft. (2.8%) 110 gals. per minute were raised at an efficiency of 30.7%; and the last installation, raising 200 gals. 1,080 ft., with 213 ft. submergence (19.6%), is operating at about 30% efficiency, using 7.25 cu. ft. free air per gallon.

* *Jour.* Chamber of Mines of Western Australia, Nov., 1910; abstract in *Eng. & Min. Jour.*, Apr. 8, 1911, p. 706.

† *Trans Am. Inst. Min. Engs.*, Feb., 1920, also personal communications to the author.

CHAPTER XXVI

COMPRESSED AIR HAULAGE

For underground transport, compressed-air and electric locomotives divide between them a broad field of operation. Both are applicable to mine service of all kinds, for hauls of a few hundred feet to several miles. But for coal mines containing fire-damp, while compressed air is perfectly safe, electric locomotives must be adopted with caution. Although, by the improvements of recent years, much has been done to prevent serious sparking at the motor, some risk still exists; and, furthermore, the possibility of strong sparking, accompanied by the momentary development of intense heat, from short circuiting or a ruptured conductor, can hardly be averted.

Besides its safety for gassy, or dry, dusty collieries, or in dry and heavily timbered workings, compressed air haulage has the following advantages: *first*, since the power is stored in the locomotive itself, the system has the maximum degree of flexibility; the locomotives can go wherever track is laid, far beyond the end of the supply-pipe line. Electric locomotives, except those having storage batteries, are dependent upon wiring, which must accompany every foot of advance.* For collieries compressed air may be used equally well for main-line haulage, and for gathering and distributing cars among the working places; *second*, compressed air costs little or nothing when not in actual use, and its full power or but a fraction of it is available at all times. In the intervals between hauls, no power is wasted, because, though the compressor may continue running, it is engaged in storing up power in the pipe-line. A minor consideration is that the exhaust of the locomotive

* "Cable-reel" or "gathering" electric locomotives are useful for very short distances only.

discharges fresh air into the workings, improving the ventilation of the mine.

At most mines compressed air is employed only for underground transport, from the stopes or breasts to the hoisting shaft. In other cases, where the mine is worked through a tunnel or adit, trains are hauled direct to the breaker, tipple, or ore-bins, on the surface. Occasionally, as at the Homestake Mine, Lead, S. D., compressed-air locomotives are used for surface transport of ore, from the crusher houses at the shaft mouths to the different stamp mills, the object being chiefly to reduce the fire risk for the wooden structures, into and near which the haulage tracks pass. For the same reasons compressed-air haulage may be installed at lumber yards, or factories containing inflammable buildings or materials.

Compressed-air locomotives were probably first used in the works of the Plymouth Cordage Co., Plymouth, Mass., about the year 1873, and in Great Britain, for mine haulage, in 1878, but the early designs were not very successful. In the United States perhaps twenty compressed-air locomotives were built previous to 1898, but since then they have been applied for a great variety of service. In general terms, the plant consists of a compressor (usually three-stage), receiver, pipe-line, charging stations, with the necessary valves, and one or more locomotives. The locomotive storage tanks are charged with a sufficient volume of high-pressure air for a round-trip run of the maximum length required, after which the locomotive returns to the nearest charging station for a fresh supply of air.

Construction and Operation of the Locomotive. For mine service the locomotive generally has one or two storage tanks, which, with the cylinders, piping, and other appurtenances, are mounted on a frame carried by 4 or 6 driving wheels.

Previous to 1908, practically all compressed-air locomotives were single-stage. Due to their greater efficiency and saving in compressed air, compound (two-stage) locomotives are now the rule, though single-stage engines are still built. Fig. 228 shows a recent design of a 4-wheel, two-stage locomotive, made in 4 sizes (Table L).

TABLE L

H. K. PORTER CO.'S TWO-STAGE COMPRESSED-AIR LOCOMOTIVES, CLASS B-P-O

Cylinders: Diameter (ins.), high pressure	$4\frac{1}{2}$	$5\frac{1}{2}$	6	7
Cylinders: Diameter (ins.), low pressure	9	11	12	14
Cylinders: Stroke (ins.)	10	10	12	14
Diameter of driving wheels (ins.)	22	23	24	26
Rigid wheel base (ft. and ins.)	2-9	2-9	3-0 or 4-0	4-0
Length over bumpers (ft. and ins.)	9 to 12-6	10 to 12-6	12 to 15-6	12 to 18-6
* Extreme width outside gage at cylinders (ins.) H P	11	$12\frac{3}{4}$	13	14
* Extreme width outside gage at cylinders (ins.) L P	$12\frac{1}{4}$	$14\frac{1}{4}$	15	$16\frac{1}{2}$
Height (see note below)				
Main reservoir capacity (cu ft.)	20 to 60	30 to 75	60 to 104	60 to 120
Main reservoir length (ft. and ins.)	6 to 9	7 to 11	9 to 12-6	9 to 15
Main reservoir diameter (ins.)	24 to 36	36 to 40	36 to 40	36 to 40
Main reservoir charging pressure (lbs. per sq in.)	700 to 1,200	700 to 1,200	700 to 1,200	700 to 1,200
Auxiliary reservoir pressure (lbs. per sq in.)	250	250	250	250
Weight in working order (lbs.)	7,000 to 10,000	10,000 to 13,000	14,000 to 17,000	18,000 to 22,000
Tractive force (lbs.)	1,450	2,200	3,000	4,400
† Hauling capacity—in tons of 2000 lbs. (exclusive of locomotive), 20 lbs. per ton rolling friction				
On absolute level	68	104	142	210
On 1% grade	32	49	67	100
On 2% grade	19	30	42	63
On 3% grade	13	21	29	45
On 5% grade	7	12	17	26
Weight per yard of lightest rail advised (lbs.)	16	16	20	30
Radius of sharpest curve advised (ft.)	15	15	20	30
Radius of sharpest curve practicable (ft.)	12	12	15	20

* Width outside of gage line may be reduced by special construction.
† These hauling capacities are maximum for conditions stated. For satisfactory operation train weights should be 50 to 90% of above figures.
NOTE.—Minimum height is dependent upon gage of track, narrow gages require more height than the wider.

Small two-stage locomotives have the high-pressure cylinder on one side, the low-pressure on the other. Larger sizes have four cylinders, a high- and low-pressure, tandem, on each side. The initial air pressure is 250 lbs., which, by expansion in the high-pressure cylinder, is reduced to 50 lbs. (for ordinary conditions the cylinder volume ratio is 1 : 4). The expanded air leaves the high-pressure cylinder at about 140° F. below atmospheric temperature (say 75°-80° below zero F.). Reheating between the cylinders is therefore necessary.

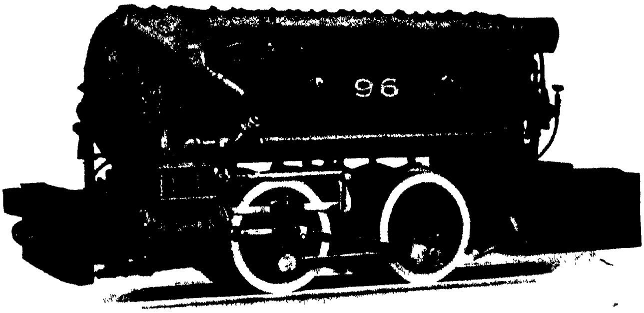


FIG. 228.—H. K. Porter Co.'s Locomotive, Class B P O (see Table L.)

Fig. 229 shows diagrammatically an efficient "atmospheric" reheater, consisting of a cylindrical shell filled with small brass or aluminum tubes, between which the exhaust air passes (see also Fig. 228, in which the reheater is shown in position). The tubes are open at both ends, the reheating medium being atmospheric air, drawn through the tubes by the ejector action of the exhaust from the low-pressure cylinder. As the volume of the high-pressure exhaust air is relatively small, its temperature is raised almost to that of the atmosphere.

A 6-wheel, single-stage locomotive, by the Baldwin Locomotive Works, is shown in Fig. 230. Dimensions: gage, 3 ft.; cylinders, 11 by 14 ins.; 2 main tanks, 22 ft. 7 ins. and 20 ft. 1 in. by 34 ins. diameter, carrying a pressure of 800 lbs.; auxiliary

tank pressure, 140 lbs.; driving wheels, 28 ins.; wheel-base, total, 6 ft. 6 ins.; total weight, 39,050 lbs., all on driving wheels. Another Baldwin locomotive, of the 4-wheel type, with 9 by 14-in. single-stage cylinders, 5-ft. 6-in. wheel-base, and weighing 24,350 lbs., is shown in Fig. 231. These builders make other

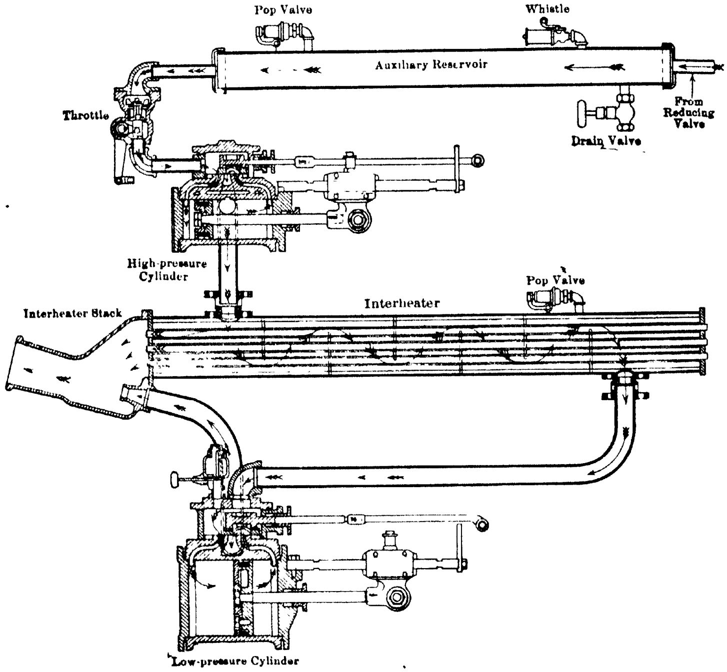


FIG. 229.—Diagram of Reheater and Cylinders for H. K. Porter Co.'s Two-Stage Locomotives (Table L).

sizes, the smallest weighing 8,000 lbs., and having 5½ by 10-in. cylinders; track gage, 36 ins.; tank pressure, 900 lbs., and working pressure 170 lbs. Working pressures of single-stage locomotives generally range from 140–180 lbs. Compressed-air mine locomotives are built also by the American Locomotive Co.

For track with sharp curves, the wheel-base must be short, say 4 ft. 6 ins. to 6 ft., for a 4-wheel engine. The height of the locomotive over all depends somewhat on the conditions existing in the mine as to thickness of vein, headroom of the haulageways, etc.; it is rarely more than 5 or 6 ft., frequently less. The length

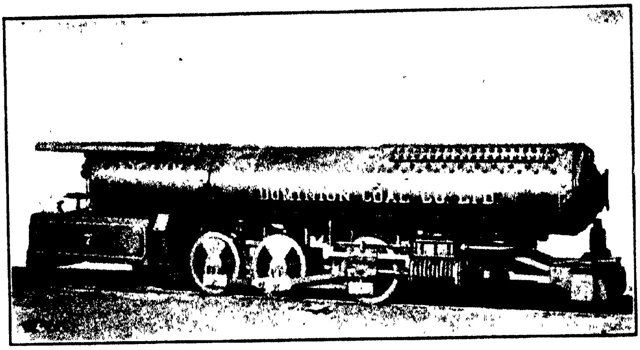


FIG. 230.—Single-Stage Locomotive, Baldwin Locomotive Works.

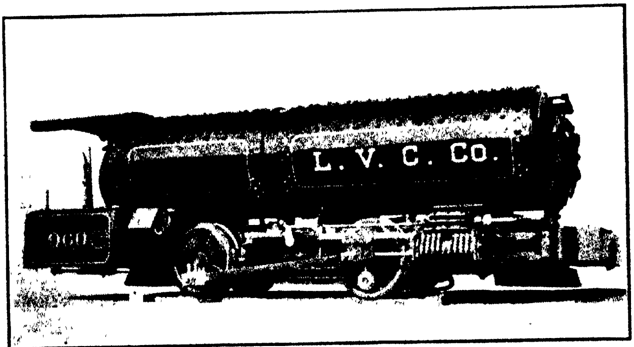


FIG. 231.—Single-Stage Locomotive, Baldwin Locomotive Works.

varies mainly according to the tank capacity required and the curvature of the gangways. It is usually 10–15 ft. for the smaller sizes, to 20 or 24 ft. for the larger. Widths, 3½–6 ft.

Fig. 232 shows an H. K. Porter Co., 4-wheel, compound locomotive, made in 4 sizes, as detailed in Table LI, and Fig. 233 a 6-wheel compound (Table LII). The hauling capacities

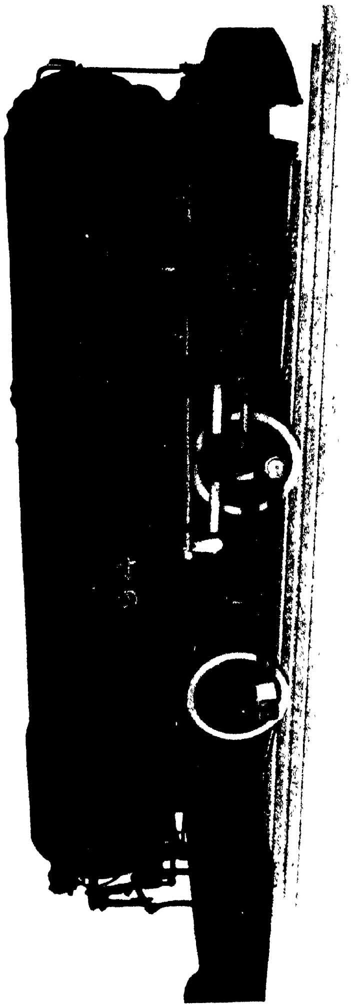


FIG. 232.—H. K. Porter Co.'s Two-Stage Locomotive, Class B-PP-O.

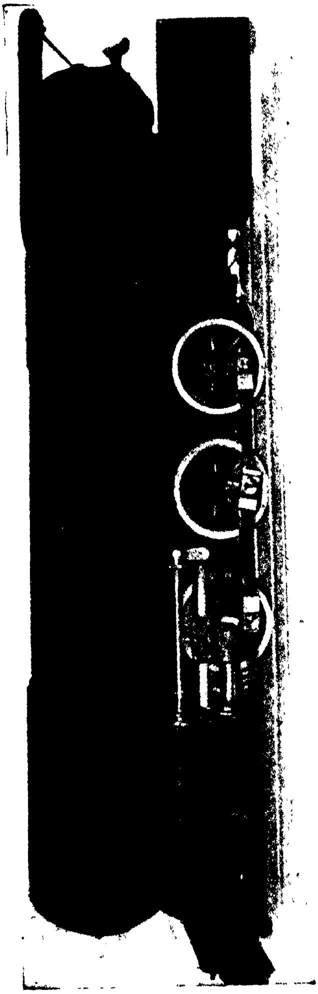


FIG. 233.—H. K. Porter Co.'s Two-Stage Locomotive, Class C-PP.

TABLE LI

H. K. PORTER CO.'S TWO-STAGE COMPRESSED-AIR LOCOMOTIVES, CLASS B-PP-O

Cylinders: Diameter (ins.), high pressure	7	$8\frac{1}{2}$	$9\frac{1}{2}$	10
Cylinders: Diameter (ins.), low pressure	14	12 and 12	14 and 14	14 and 14
Cylinders: Stroke (ins.)	14	14	14	14
Diameter of driving wheels (ins.)	26	26	26	26
Rigid wheel-base (ft. and ins.)	4-0	4-6	4-6	5-0
Length over bumpers (ft.)	14 to 20	16 to 20	18 to 21	20 to 23
* Extreme width outside gage at cylinders (ins.) H.P.	14	$15\frac{1}{2}$	$16\frac{1}{8}$	$17\frac{3}{4}$
* Extreme width outside gage at cylinders (ins.) L.P.	$16\frac{1}{2}$	$16\frac{3}{4}$	$17\frac{7}{8}$	$17\frac{7}{8}$
Extreme width across main reservoir tanks (ft. and ins.)	5-5 to 6-5	5-5 to 6-5	5-10 to 6-10	5-10 to 6-10
* Height above rail (ft. and ins.)	5-0 to 5-10	5-0 to 5-10	5-0 to 5-10	5-5 to 5-10
Main reservoir, 2 tanks, capacity (cu. ft.)	97 to 160	114 to 210	160 to 275	160 to 290
Main reservoir lengths (ft.)	10 to 18	12 to 18	14 to 20	16 to 21
Main reservoir diameters (ins.)	30 to 36	30 to 36	$32\frac{1}{2}$ to $38\frac{1}{2}$	$32\frac{1}{2}$ to $38\frac{1}{2}$
Main reservoir charging pressure (lbs. per sq. in.)	700 to 1,200	700 to 1,200	700 to 1,200	700 to 1,200
Auxiliary reservoir pressure (lbs. per sq. in.)	250	250	250	250
Weight in working order (lbs.)	20,000 to 27,000	28,000 to 36,000	32 000 to 42,000	43,000 to 46,000
Tractive force (lbs.)	4 400	6,400	8,000	9,000
Hauling capacity—in tons of 2,000 lbs. (exclusive of locomotive), 20 lbs. per ton rolling friction:				
On absolute level	206	305	380	437
On 1% grade	06	145	180	207
On 2% grade	59	92	113	130
On 3% grade	31	65	80	02
On 5% grade	22	38	46	53
Weight per yard of lightest rail advised (lbs.)	30	40	45	50
Radius of sharpest curve advised (ft.)	30	35	40	50
Radius of sharpest curve practicable (ft.)	16	18	20	25

* This may be decreased in special cases.

TABLE LII

H. K. PORTER CO.'S TWO-STAGE COMPRESSED-AIR LOCOMOTIVES, CLASS C-PP

Cylinders: Diameter (ins.) H.P.	7	8½	9½	10
Cylinders: Diameter (ins.), L.P.	14	12 and 12	14 and 14	14 and 14
Cylinders: Stroke (ins.)	14	14	14	14
Diameter of driving wheels (ins.)	26	26	26	26
Rigid wheel-base (ft. and ins.)	5-6	5-6	5-6	6-6
Length over bumpers (ft.)	14 to 20	16 to 20	18 to 21	20 to 23
* Extreme width outside gage at cylinders (ins.) H.P.	14	15½	16⅛	17⅛
* Extreme width outside gage at cylinders (ins.) L.P.	16½	16⅜	17⅞	17⅞
Extreme width across main reservoir tanks (ft. and ins.)	5-5 to 6-5	5-5 to 6-5	5-10 to 6-10	5-10 to 6-10
Height above rail (ft. and ins.)	5-0 to 5-10	5-0 to 5-10	5-0 to 5-10	5-0 to 5-10
Main reservoir (2 tanks) capacity (cu ft.)	97 to 160	114 to 210	160 to 275	160 to 290
Main reservoir lengths (ft.)	10 to 18	12 to 18	14 to 20	16 to 21
Main reservoir diameters (ins.)	30 to 36	30 to 36	32½ to 38½	32½ to 38½
Main reservoir charging pressure (lbs. per sq in.)	700 to 1.200	700 to 1.200	700 to 1.200	700 to 1,200
Auxiliary reservoir pressure (lbs. per sq in.)	250	250	250	250
Weight in working order (lbs.)	20,000 to 27.000	28.000 to 36.000	32,000 to 42,000	43,000 to 46,000
Tractive force (lbs.)	4.400	6.400	8,000	9.200
Hauling capacity	The figures in Table LI apply here also			
Weight per yard of lightest rail advised (lbs.)	25	30	35	40
Radius of sharpest curve advised (ft.)	50	50	50	70
Radius of sharpest curve practicable (ft.)	30	30	30	50

* This may be decreased in special cases.

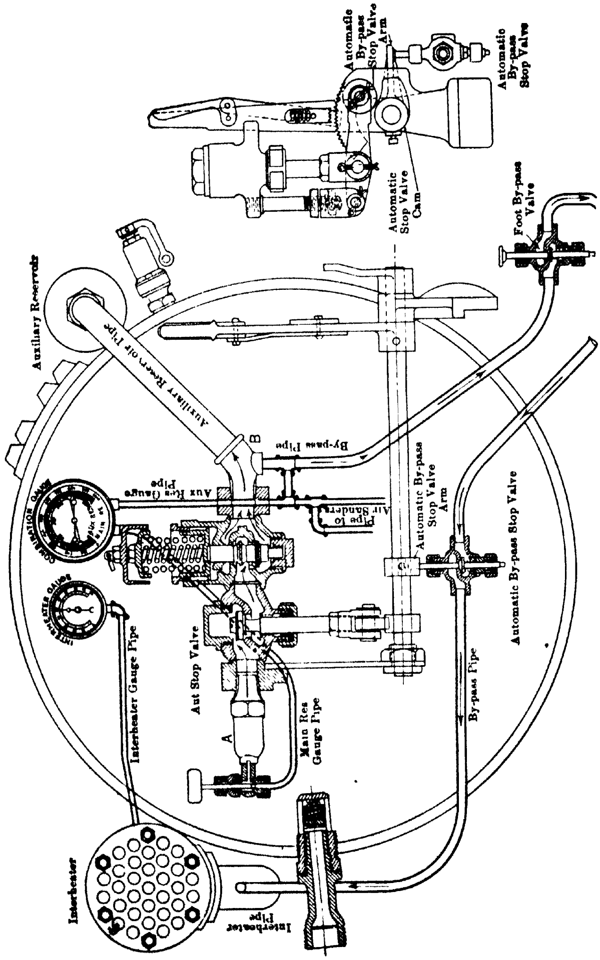


FIG. 235.—Diagram of Air Piping, Interheater, Valves, etc., on Rear End of Two-Stage Locomotive (H. K. Porter Co.)

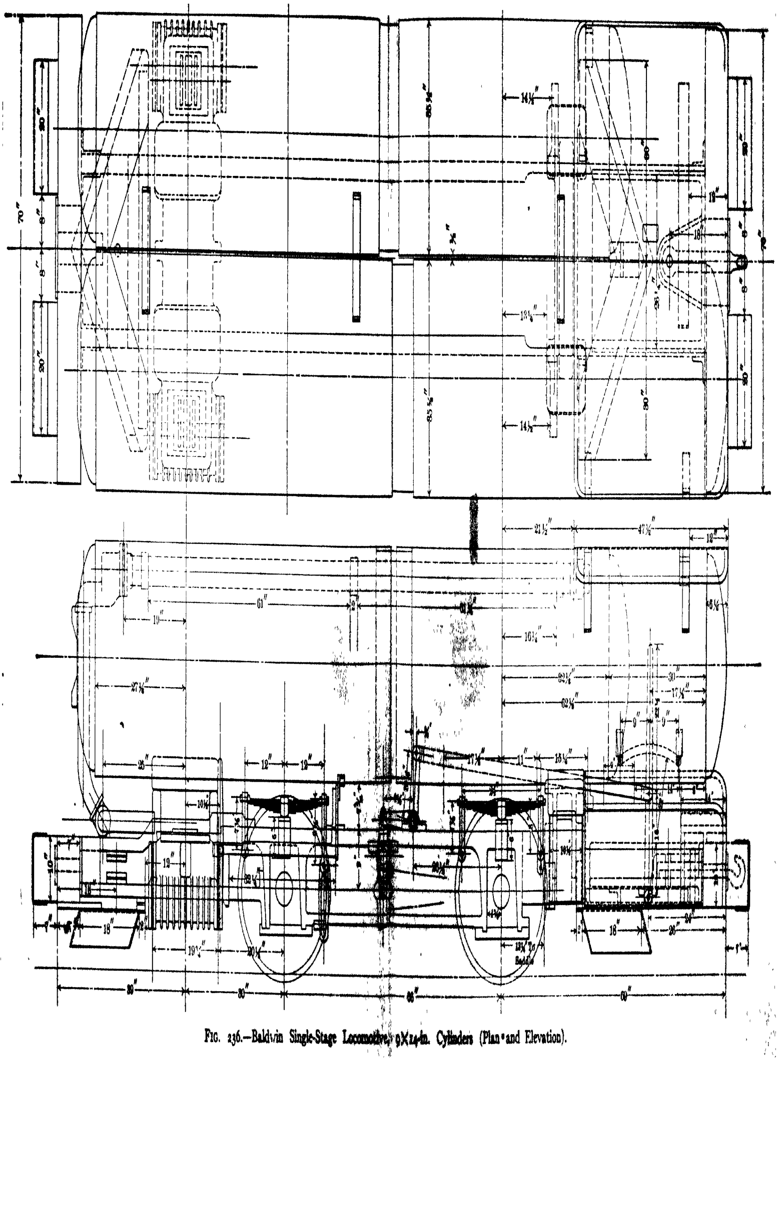


FIG. 236.—Baldwin Single-Stage Locomotive, 9×24-in. Cylinders (Plan and Elevation).

pressures of 2,000–2,200 lbs. per sq. in. These high pressures are unnecessary for ordinary haulage service.

From the main tank the air passes into a small auxiliary reservoir and thence to the cylinders. This reservoir is usually a section of wrought-iron pipe, 4–9 ins. diameter and 6–15 ft. long, laid alongside the main tank. The pressure in it is adjusted by an automatic reducing valve to the requirements of the engine—usually 140–180 lbs. for single-stage, and 225–250 lbs. for two-stage locomotives, depending on the size of cylinders and the power required. On the locomotives of the H. K. Porter Co., the reducing valve is a double-seated balanced valve, operated by a small piston. The air pressure in the

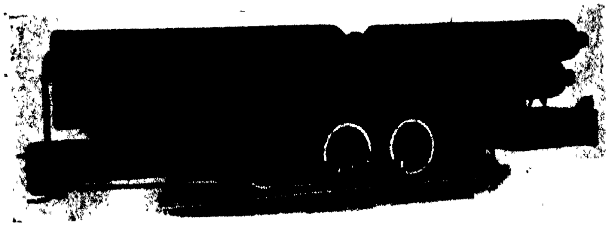


FIG. 2 7.—H K. Porter Co.'s Locomotive, Class C-5PB-O.

reservoir acts on one side of the piston, tending to close the valve. This action is opposed by an external spring, adjusted to keep the valve open until normal working pressure is reached in the reservoir. Then the valve is closed by the air pressure, against the spring resistance. To provide for the case when the locomotive is using no air (as on a down grade or when at rest), a single-seated supplementary valve is placed in the pipe between the reducing valve and the main tank. This valve is controlled by the throttle lever; being open when the throttle is open, otherwise closed by the air pressure. Having the two valves, leakage from main tanks to auxiliary reservoir is avoided and a close regulation secured.

From the auxiliary reservoir the air passes to the cylinders through a balanced throttle valve. This maintains a constant

working pressure, suited to the needs of the locomotive, prevents waste of air likely to ensue if air at full tank pressure were admitted to the cylinders, and makes the locomotive more manageable. In starting a heavy load excessive slipping of the drivers is avoided, and with light loads the reducing valve may be regulated for any desired pressure. Toward the end of the trip, when the pressure in the main tanks falls to that in the auxiliary, the cylinders take air directly from the former, and the locomotive will continue to run as long as the pressure remains sufficient. For long hauls, or when the cross-sectional dimensions or sharp curves, or both, of the haulage-ways do not permit the use of tanks of great length or large diameter, a tender carrying a supplementary tank may be employed (Figs. 238 and 246).

FIG. 238.—H. K. Porter Co.'s Locomotive, Class B-PP-T (Cylinders are inside of the frame).

For small, single-stage locomotives, the air is sometimes admitted to the cylinders throughout nearly full stroke, and consequently, as the exhaust is at high pressure, the efficiency is low. This practice is due to the tendency to use as small a locomotive as possible, on account of the limited headroom and narrow, crooked gangways so common in mines. Better results are obtained by working with a cutoff, and increas-

ing the size of the cylinders and the weight on the drivers. In using air expansively, as can be done with properly proportioned cylinders, there should be no trouble from freezing of moisture. Although expansion produces a low cylinder temperature, yet, as the initial pressure is so much higher than is employed for other compressed-air machinery, the expanded air is relatively dry, and the force of the exhaust is sufficient to keep the ports clear of ice. To this end the ports should be large, straight, and short. The cylinders are not lagged with non-conducting covering, as is so necessary for steam cylinders, to minimize condensation. By exposing their surface to the warm air of the mine, some heat is absorbed. Occasionally the exterior surface of the cylinders is cast with deep corrugations, to present a large area to the warm surrrounding air (Figs. 230, 231, and 236). The cylinders are provided with slide valves; piston valves, like those used in steam locomotives, would leak more because of the dryness of the air.

On account of the cold produced by reducing the pressure between the main tank and auxiliary reservoir, and to increase efficiency of operation, reheating is advantageous (though not essential) for single-stage locomotives. It is best done by heating the auxiliary reservoir. If steam be available in the mine, a little steam and hot water may be injected into the reservoir each time the locomotive is charged. Or, in mines free from fire-damp, a small reheater, burning oil or coke, may be carried on the locomotive. When the air is reheated water should always be kept in the small tank; the moisture from it, passing with the air into the cylinders, assists in lubricating the valves and pistons. For two-stage locomotives, reheating is unnecessary, except between the cylinders (see Fig. 229 and accompanying description, p. 435).

Pipe-Line and Charging Stations. The required capacity of the plant depends on the length of haul and size of locomotives, as influenced by the daily output, weight of trains, and track gradients. For short hauls, the pipe-line is sometimes omitted, the locomotive returning to the compressor receiver to be recharged. Usually a pipe-line is carried underground, with

charging stations at one or more points, located according to the haulage distances and the capacity of the locomotive tanks. The innermost station, farthest from the compressor, must be at a point from which the locomotive can reach the end of its trip and return for recharging. For long hauls, heavy traffic, or adverse gradients, several charging stations may be required. Inside receivers are unnecessary, unless the diameter of the pipe-line is too small. The pipe-line itself acts as a storage reservoir, and should be of a diameter which, in proportion to its length, will furnish a cubic capacity sufficient to charge the locomotive tank quickly and without excessive drop in pressure. That is, the pressure in the tank and pipe-line on equalizing should not fall much below the pressure which the locomotive is designed to carry. To this end, the volume of pipe-line storage should be at least three times the tank capacity of the locomotive. To determine the necessary pipe-line capacity, several variables must be harmonized, as follows:*

V = storage volume required, cu.ft.;
v = volume of locomotive tank, cu.ft.;
P = pipe-line pressure, lbs. per sq. in. (usually 900–1,200 lbs.);
p = desired pressure in locomotive tank, lbs. per sq. in. (700–900 lbs.);
p' = residual pressure in locomotive tank, just before recharging, lbs. per sq. in.

$$\text{Then: } V(P-p)=v\,(p-p'), \text{ or } V=\frac{v(p-p')}{P-p}$$

For example, let $P=900$ lbs., $p=750$ lbs., $p'=125$ lbs., and $v=100$ cu.ft., from which:

$$V=\frac{100(750-125)}{900-750}=416.6 \text{ cu.ft.}$$

By transposition, the same formula serves for finding the pipe-line pressure required for a given tank pressure. When there are several locomotives, it is rarely necessary to design the pipe-line for charging more than one at a time. If the volu-

* H. K. Porter Co. "Modern Compressed-Air Locomotives," 1916.

metric capacity of the pipe-line be ample, the drop in gage pressure on charging is soon recovered by the compressor, which, except in plants operating a single locomotive, is kept in nearly constant operation. If more locomotives are added after the original installation of the system, the same pipe-line may still serve, provided the compressor is able to charge it to full pressure at shorter intervals.

The piping, which is generally from 3–5 in., should be of the best material, lap-welded, and with sleeve joints made with the utmost care to prevent leakage. To stop leaks, the sleeves should have annular grooves (counter-bores) at each end, into which lead calking is driven if required. About every 300 ft. a pair of flanges should be placed in the line, and valves at suitable places, for convenience in making repairs or extensions. There should also be a valve between the compressor and pipe-line, so that the compressor can be inspected or repaired without wasting the air in the pipe. The pipe should not be buried, but carried along one side of the haulageway, either on the floor or on brackets, so that leaks will at once attract attention. An occasional bend in the pipe-line is advantageous in permitting free expansion and contraction, but bends should not be too numerous, as they involve more joints and greater possibility of leakage. Pipe-lines should be painted with some non-corrosive coating.

Charging Stations. Fig. 239 shows one design, consisting of a heavy tee inserted in the air main, with a flexible connection for coupling to the locomotive. The connecting pipe comprises a vertical, rigid branch, with a 1½-in. gate-valve, and a short horizontal pipe, with a union and a ball-and-socket joint. A further flexible connection, with 2 ball joints, serves for coupling to the locomotive tank. Thus, for charging, the locomotive need not be spotted accurately on its track, but has a foot or two leeway. When not in use, the flexible connection is swung aside.

After coupling to the locomotive, the gate-valve is opened, whereupon the air pressure immediately forces together the parts of the ball-and-socket joints and makes a tight connec-

tion. When equilibrium is established between the pressures in the pipe-line and tank, the gate-valve is closed. To break the coupling, the compressed air remaining in the connect-

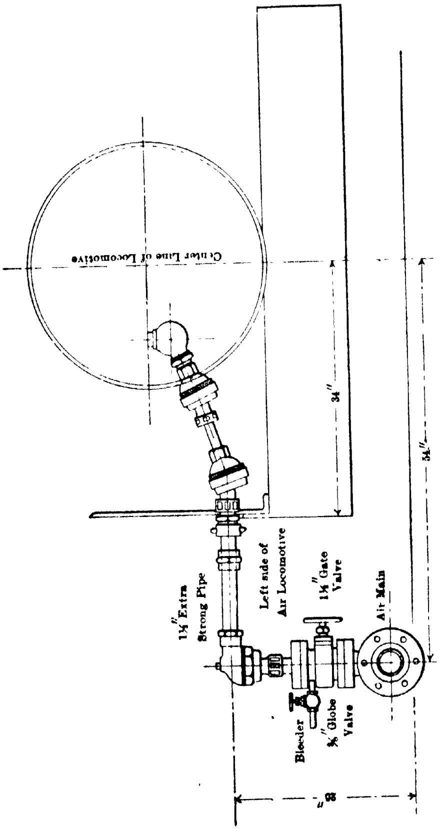


FIG. 239.—Compressed-Air Locomotive Charging Station.

ing pipe, between the gate-valve and the locomotive check-valve, must first be released. This is done by opening a small "bleeder valve," placed just above the gate-valve. The joints then loosen and are readily manipulated. The actual time

occupied in charging is usually about three-quarters of a minute, but, including stopping the locomotive and making the connection, 1½–2 mins. may be allowed. Charging may often be done while shifting cars and making up trains. Fig. 240 shows another form of charging station.

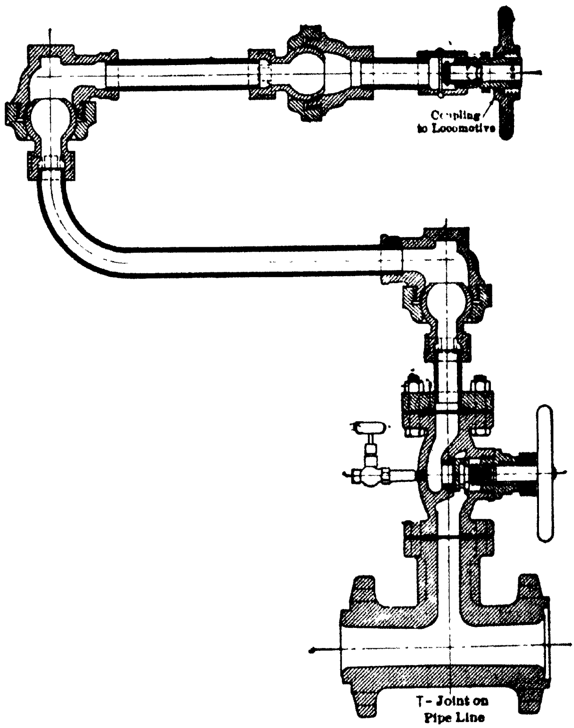


FIG. 240.—Charging Station (H. K. Porter Co.).

Calculation of Motive Power. Several factors must be known, *viz.*, the tractive resistance per ton of the loaded cars on a level, resistances due to gradients and curves, weight of empty and of loaded cars, and number of cars to be hauled per train. The values of these factors are readily ascertained, with

exception of the resistances due to curvature of track and character of roadbed. The former has been determined experimentally for ordinary surface railways, but mine track is apt to be roughly laid, with curves of varying and irregular radius, and the elevation of the outer rail improperly adjusted.

The average tractive force required per ton depends on the condition of the track and roadbed, and the character and state of repair of the running gear of the cars. On level mine track the coefficient of rolling friction is usually from 30–40 lbs. per ton, though it may be considerably higher on poorly laid or light track. With mine track in exceptionally good condition, this coefficient may be as low as 20 lbs. The grade resistance is 20 lbs. per short ton, for each 1% of grade. The distribution of grades is often such that the maximum load is not the resistance of the loaded trains, which are usually hauled on slight down grades, but that of the return trains of empty cars on adverse gradients. For the most economical work, gradients should not exceed ½–¾ of 1% in favor of the loaded trains. With ordinary track and rolling stock, and a grade of 5–6 ins. per 100 ft., the coefficient of rolling friction is nearly the same for a loaded train hauled down as for an empty train of the same number of cars hauled up the grade. Heavier adverse grades are often necessary (2½%–3% or more), but they should be avoided, because the maximum tractive force of a locomotive falls off rapidly. On a 2½% adverse grade the locomotive can haul only about 4 times its own weight, even if the track be not slippery. Grades should be reduced on curves. Colliery cars carrying 2½–3½ tons weigh 1,800 –2,300 lbs., while those used in metalliferous mines, for mechanical haulage, weigh 1,000 and 2,000 lbs. Having ascertained the values of the different factors, the proper allowance of reserve power, in terms of volume and pressure of air, to cover indeterminate resistances due to local imperfections of track and rolling stock, is a matter of judgment and experience.

With a given air pressure, the capacity of the locomotive tanks depends primarily on the length of round trip to be made with a single charge of air. When this distance is 1–1½ miles,

the tank capacity is generally from 50 to 150 cu.ft., according to the load; which, in turn, together with the track and grade resistances, governs the cylinder dimensions. Cylinders of 5 by 10 ins. up to 9 by 14 ins. are common, the larger sizes being for heavy work in collieries or on surface. For runs exceeding 1½ miles, it is often desirable to increase the air pressure, rather than the tank capacity. Another plan is to provide a tender, carrying an auxiliary tank (Figs. 238 and 246).

Having determined the total foot-pounds of work to be done with a single charge of air, on a run of the maximum length, specifications may be obtained from the builders for a locomotive of suitable weight, gage, wheel-base, tank capacity, and cylinder dimensions.

Compressors for Charging Locomotives are three- or four-stage. The air cylinders of the higher stages are single-acting. Fig. 241 shows a three-stage locomotive charger built by the Norwalk Iron Works Co., for pressures up to 1,000 or 1,200 lbs. The air passes from the low-pressure cylinder to the lower of the two intercoolers and, thence to the intermediate cylinder. From the latter the air goes through the vertical pipe to the upper intercooler, and thence through the inclined pipe to the high-pressure cylinder, from which the compressed air is delivered to the receiver through the connection indicated under the outer end of the cylinder.

The air end of a three-stage locomotive charger, by the Ingersoll-Rand Co., is shown in Fig. 242. The high pressure intercooler is in the lower right-hand corner of the cut. Figs. 243 and 244 illustrate a duplex, four-stage compressor; in Fig. 243 are the intake and first intermediate cylinders, and in Fig. 244 the second intermediate and high-pressure cylinders. Fig. 245 shows the entire compressor.

The pistons of the high-pressure cylinders are solid rams, with a series of packing rings. These, with the high-pressure valves, must be made with especial care, to prevent the serious effects of leakage of high-pressure air. Locomotive chargers are also built by the Sullivan Machinery Co. and others.

When the mine is already provided with an ordinary low-

pressure air plant, for machine drills, etc., and which has some surplus capacity, a two-stage charging compressor may be installed, to take air from the low-pressure system and bring it up to the tension required for the locomotives. Some reduction

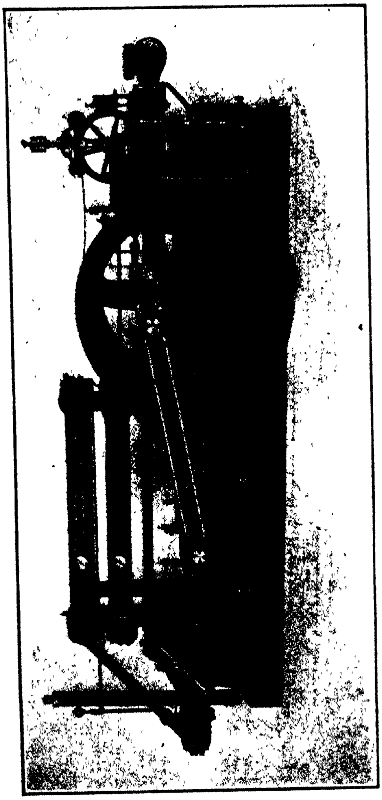


FIG. 241.—Norwalk Locomotive-Charging Compressor.

in cost of plant may thus be effected, but care must be exercised in making such a combination, and it is not advisable. If the quantity of air produced by the low-pressure system should at times be insufficient to furnish the necessary excess for the

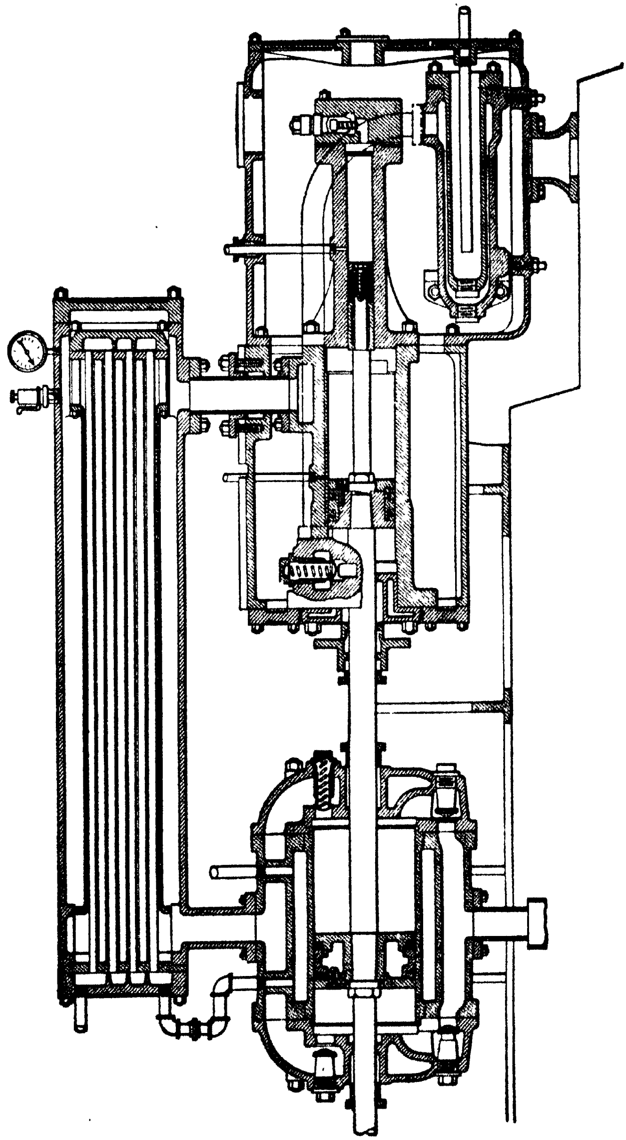
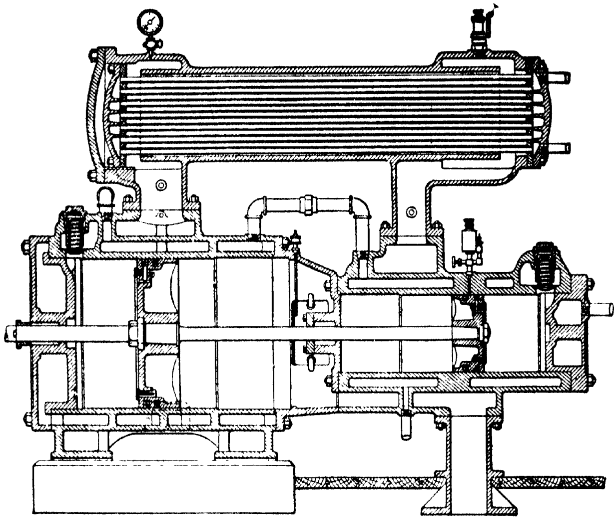
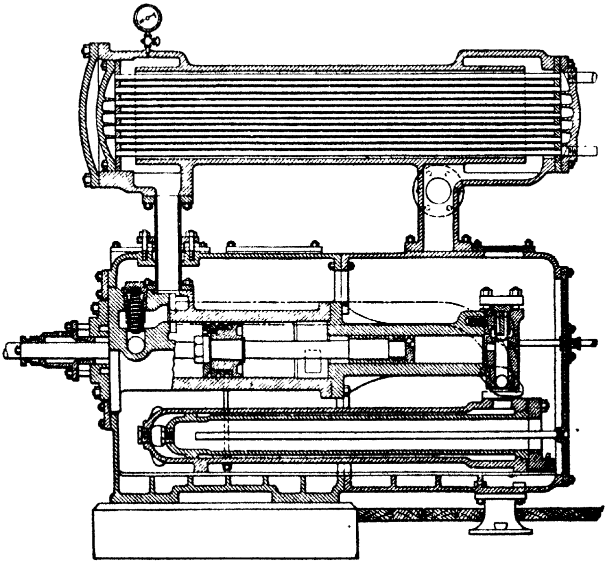


FIG. 242.—Air End of Ingersoll-Rand Three-Stage Locomotive Charger.





FIGS. 243 and 244.—Ingersoll-Rand Four-Stage Locomotive Charger.

locomotive charger, the latter might have to compress from too low an initial pressure. There would be excessive development of heat which might raise the cylinder temperature to the flashing-point of the oil, thus causing an explosion (Chap. XIV).

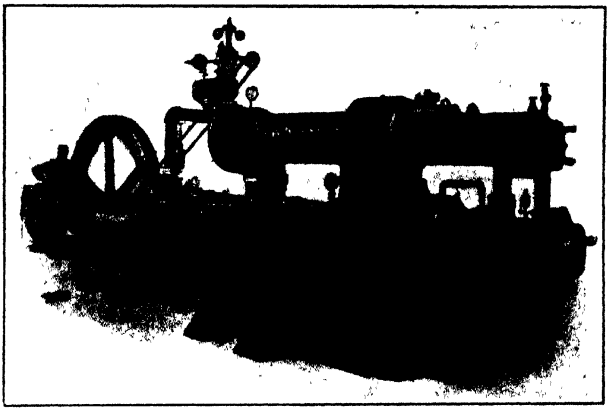


FIG. 245.—Ingersoll-Rand Four-Stage Compressor.

Capacity of the Charging Compressor depends on the pipeline pressure, number of locomotives to be operated, cubic contents of the locomotive tanks, pressure carried by the system, and the relation between charging periods.

Let C = compressor capacity required, cu.ft. free air per min.;

L = locomotive-tank capacity, cu.ft. free air per min.;

N = number of charges required in any given time, T.

Hence the equation: $C=\frac{NL}{T}$

For example, if N = 3, L = 5,200 (corresponding to 100 cu. ft. of air at 750 lbs. gage pressure), and T = 60 minutes:

$$C=\frac{3\times 5,200}{60}=260 \text{ cu.ft. free air per min.}$$

When the locomotives are charged at approximately equal intervals of time, a single application of the above formula will be sufficient. Otherwise, calculations are required to determine

the maximum and minimum rates of consumption of air. For plants installed at an altitude above sea-level, allowance must be made for decreased output (Chap. XIII).

Examples of Compressed-Air Haulage Plants.

1. At the Buck Mountain Colliery, Penn., are two 8-ton H. K. Porter locomotives, each with 2 tanks, 15 and 17 ft. long, having a combined capacity of 130 cu.ft. of air at 550 lbs. pressure. Cylinders, 7 by 14 ins.; wheel-base, 5 ft. 3 ins.; height, 5 ft. 2 ins.; length over all, 19 ft.; gage of track, 42 ins. A round trip of 5,100 ft. is made in 30-40 mins., or 2,500 ft. in 12-15 mins, with 12-car trains, on grades of ½-4½%, averaging ¾ of 1% in favor of the load. One locomotive delivers 150 cars per 10 hours, doing the work formerly done by 15 mules. Weight of cars, 3,400 lbs. empty, and 10,400 lbs. loaded. A three-stage Norwalk compressor supplies 375 cu.ft. free air per min., at 700 lbs. gage. Pipe-line, 4 ins. diameter and 9,600 ft. long; storage capacity, 800 cu.ft.

Average cost per ton-mile: 1.875 cents for the gross weight hauled, or 3.77 cents for net weight of coal. The cost for mule haulage under the same conditions was formerly 3.94 and 7.92 cents, respectively.

The cost of this plant was as follows:

Two locomotives		$5,505
Air line: 9,647 ft of 4 in pipe	$2,894	
Six charging stations	360	
Fittings and valves	382	
Labor cost for erection	998	
		4,634
Compressor	$2,880	
Sundries and erection	246	
Compressor house	256	
Steam line to compressor	152	3,534
Total cost		$13,673

2. Empire Mine, Grass Valley, Cal. Several small locomotives, built by Edward A. Rix, are employed in the deep levels, for hauling trains of 5 cars, each carrying 1 ton. Maximum distance per round trip, about 5,000 ft. Locomotive tank, 36 ins. diameter by 48 ins. long; pressure, 500 lbs; dimensions

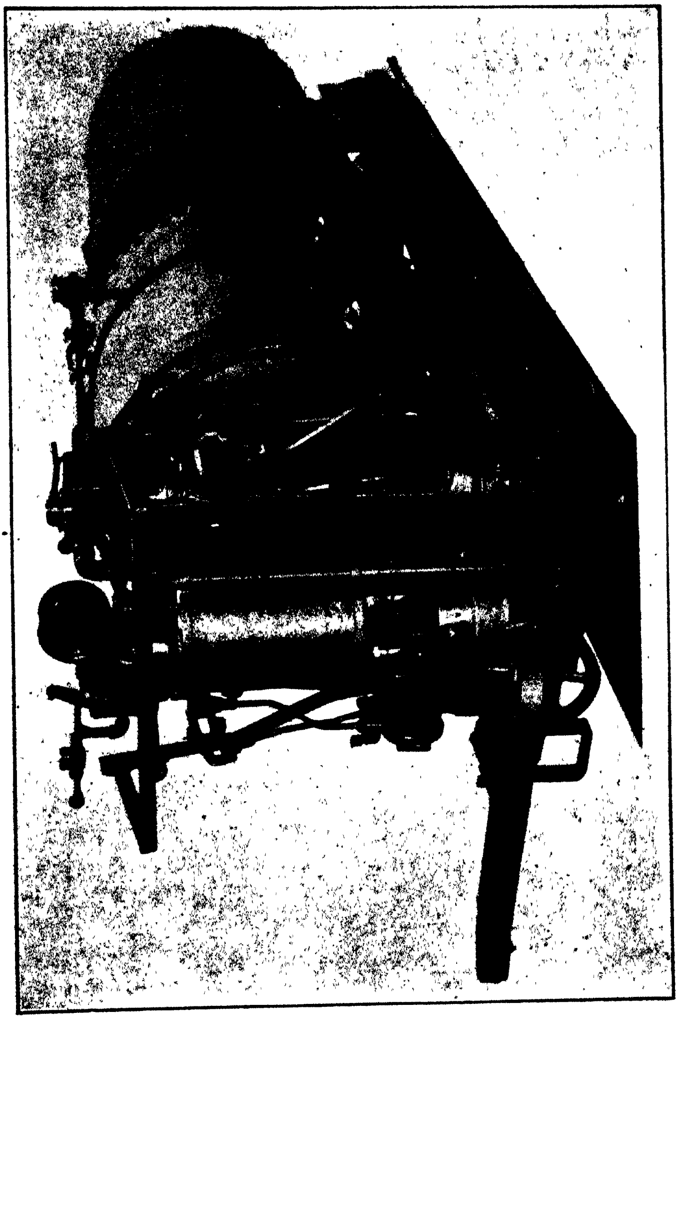


FIG. 246.—Compressed-Air Locomotive, Empire Mine, Cal.

of locomotive, 5 ft. long by 30 ins. wide by 52 ins. high; gage of track, 18 ins. One locomotive (Fig. 246) is operated by a pair of vertical engines, a chain and sprocket drive connecting the crank-shaft with the rear axle. There are 2 tandem tanks, one of them carried on a tender. A reheater (Primus kerosene burner) reheats the air after its pressure has been reduced in the auxiliary reservoir. Mr. Rix has built 3 similar locomotives, but with a single, larger tank, for a 3-mile tunnel, near San Francisco. They carry 1,000 lbs. tank pressure, the working pressure being 100 lbs.; each makes a 2-mile round trip, at 6–7 miles per hour.*

3. The Peerless Colliery, Vivian, West Va., operated for years several H. K. Porter locomotives, with 5 by 10-in. cylinders and weighing 10,000 lbs. Over-all dimensions: 10 ft. 5½ ins. long by 5 ft. 8 ins. wide by 4 ft. 5 ins. high. Four driving wheels, 23 ins. diameter; gage, 44 ins. Capacity of storage tank, 47 cu.ft.; pressure, 535 lbs.; charging time, 20 seconds; working pressure, 125 lbs. Pipe-line, 3 ins. diameter, with a capacity of 242 cu.ft. Line pressure, 735 lbs. Trains consist of 6 cars, each weighing, loaded, 8,500 lbs. Grades range from level to 2½%, generally in favor of the load. Curves from rooms to haulageways, 23 ft. radius, though the locomotives can work on curves of 15-ft. radius. Maximum round trip, 9,000 ft.; maximum speed, 10–12 miles per hour. Cost of each locomotive, $1,800.

4. The following data, on one of the plants of the Philadelphia & Reading Coal & Iron Co., were compiled by Mr. G. Clemens, a division engineer of the Company:

a. Shaft level—1 locomotive.

Round trip, 5,200 ft.; grades $\frac{4}{10}$ to $\frac{8}{10}$ of 1%, all in favor of load; charging station at each end of run; gage of track, 44 ins.; weight of cars, empty, 3,300 lbs., loaded, 8,800 lbs.; 15–38 cars per trip; total output, 600 cars per 10 hours. Round-trip time, 12 min.; charging time, 1 min. A round trip and a half can be made with one charging.

b. Slope level—1 locomotive.

* *Compressed Air Magazine*, Feb., 1908, p. 4747.

Length of haul, 3,200 ft., of which 700 ft. is on an adverse grade of $4\frac{1}{10}$ to $5\frac{1}{2}$%. Grade of main gangway, $\frac{4}{10}$ to $\frac{8}{10}$ of 1%, in favor of load. Trains of 10 cars are hauled on main gangway, and 4 cars on the slope; weights of cars same as above.

Locomotive-tank pressure at start, 600 lbs.; at end of trip, 200 lbs. Average working pressure, 180 lbs. The cost of the plant was as follows:

One Norwalk 3-stage compressor, erected	$5,180 74
Pipe-line, 4,200 ft, 5 in, including 3 charging stations.	2,951 06
Two Baldwin compressed-air locomotives and fittings	4,904 33
Alterations in gangways to adapt them to locomotive haulage	665 17
Total cost.	$13,701 30
Daily operating cost, for 180 days in the year	$14 69
Fixed charges, depreciation, repairs, etc, figured at 10 per cent, together with cost of steam power	9 00
Total running expenses per day	$23 69
Haulage cost per car, at 600 cars per day	3 6 cents
Previous cost of mule haulage per car	5 1 "
Saving per year, about	$1,800 00

5. At the Wilson Colliery, of the D. & H. Coal Co., a large locomotive was installed by the Dickson Manufacturing Co., having six 26-in. drivers; wheel-base, 7 ft.; cylinders, 9 by 14 ins.; gage of track, 30 ins. The locomotive carries two tanks, 18 ft. 6 ins. and 15 ft. 6 ins. by 30 ins. diameter; capacity, 160 cu.ft. of air at 600 lbs. Pipe-line, 4,100 ft. long; pressure, 700 lbs. Total charging time, 1 min. 25 seconds. After reduction to 125 lbs. working pressure, the air is reheated. Trains are usually of 30 cars, each weighing loaded, 5,850 lbs. (the locomotive has a capacity of 50 cars). Grades from $\frac{3}{4}$ of 1% against the load, to 1% in favor. Round-trip time, for 8,200 ft. plus a switching distance of 800 ft., 16 min. Cost of haulage per ton-mile, gross, about $1\frac{1}{4}$ cents.

6. The Anaconda Copper Mine, Butte, Mont., has a number of compressed-air locomotives with 5 by 10-in. cylinders and

weighing 10,000 lbs. Over-all dimensions: height, 58 ins.; width, 58 ins.; length, 10 ft. 4½ ins.; 4 driving wheels, 23 ins. diameter; wheel-base, 36 ins., designed for curves of 12-ft. radius. Capacity of main tank, 47 cu.ft.; pressure, 550 lbs.; working pressure, 125 lbs.; charging time, 60 seconds. Length of haul, 2,400 ft. round trip; load, 6 cars, weighing loaded 3,450 lbs. each; track nearly level. The locomotives can make 2 round trips, or 4,800 ft. on 1 charge, with cold air; by reheating with hot water, 3 round trips can be made.

At the Anaconda reduction works, there are 13 H. K. Porter locomotives, employed in handling the products between the different divisions of the plant; length of haul, 1,000–7,000 ft. Twelve locomotives have the following dimensions; weight, 26,000 lbs.; cylinders, 9½ by 14 ins.; driving wheels, 28 ins.; wheel-base, 54 ins.; main tanks, 132 cu.ft.; drawbar pull, 5,700 lbs. Another locomotive weighs 42,000 lbs.; cylinders, 12 by 18 ins.; driving wheels, 36 ins.; wheel-base, 60 ins.; main tanks, 218 cu.ft.; drawbar pull, 9,180 lbs. Tank pressure, 700–800 lbs.; working pressure, 150 lbs.

7. The Homestake Mining Co., Lead, S. D., employs underground 10 H. K. Porter locomotives, weighing 9,500 lbs. and measuring over all, 4 ft. 11 ins. high by 3 ft. 3½ ins. wide by 10 ft. 6 ins. long. Gage of track, 18 ins. They have a detachable rear end (as in Fig. 234) to permit of transferring them from level to level, on a cage with a 9-ft. platform.

8. At the Aragon Iron Mine, Norway, Mich., is an H. K. Porter locomotive. Weight, 7 tons; height, 5 ft. 2 ins. ; width, 4 ft. 2 ins. ; length, 12 ft. over all. Four 24-in. drivers; wheel-base, 48 ins.; gage, 22½ ins.; cylinders, 7 by 12 ins.; tank pressure, 700 lbs.; working pressure, 140 lbs.; charging time, 30–60 seconds. Haulage distance, 1,200–4,000 ft.; pipe-line, 1,800 ft. Locomotive hauls four 20-car trains per 10 hours, from each of 10 loading places. Weight of loaded train, including locomotive, 43 tons; weight empty train, 18 tons. At the compressor are 2 receivers, each 3 by 17 ft.

9. Compressed-air haulage plant at No. 6 Colliery, Susquehanna Coal Co., Glen Lyon, Penn. Following is an abstract

of tests by J. H. Bowden and R. V. Norris.* Though the plant is old, the results are useful.

Compressor: Norwalk, three-stage; steam cylinder, 20 by 24 ins.; air cylinders, 12½, 9½ and 5 ins. by 24 ins.; capacity, at 100 revolutions, 296 cu.ft. free air per min., compressed to 600 lbs. Main pipe-line at No. 6 shaft, 4,380 ft. long, 5 ins. diameter, with 5 charging stations, and capacity of 608 cu.ft. Branch line, in No. 6 slope, 3,100 ft. long, 3 ins. diameter, with 3 charging stations, and capacity of 159 cu.ft.

Locomotives: two, by H. K. Porter Co.; weight, 8 tons; tank capacity, 130 cu.ft.; pressure, 550 lbs., reduced to 160 lbs. in an 8-in. auxiliary reservoir of 4.2 cu.ft. capacity. Cylinders, 7 by 14 ins.; four 24-in. drivers.

At No. 6 shaft the run averages 4,000 ft. each way, on ½–2¾% grades, averaging about 1%, in favor of the load. Run at No. 6 slope averages 2,100 ft., with nearly the same grades. Cars weigh 2,800 lbs. empty, and about 9,800 lbs. loaded; trains, 12–20 cars. The shaft locomotive hauls about 355, and the slope locomotive 320 cars, per 10 hours, doing the work of 32 mules. Tests on the compressor showed 150 indicated H.P. at 131 revs., compressing 387.8 cu.ft. free air per min.

The combined capacity of both pipe-lines is 767 cu.ft., which, at 600 lbs. gage pressure, is equivalent to 32,500 cu.ft. free air. Capacity of locomotive main and auxiliary tanks, 134.6 cu.ft.; at 508 lbs. (at which pressure the tanks equalize with the mains, the initial pressure being 600 lbs.), this is equivalent to 4,845 cu.ft. free air. In standing 12 hours, the pipe-line pressure falls to 350 lbs.; hence the loss per hour from leakage is 974 cu.ft. free air, or 4.18% of total air compressed.

Average volume free air used by both locomotives per ton-mile: gross, 100 cu.ft.; net, 180 cu.ft. Another test showed a total consumption of 223,020 cu.ft. free air, for hauling 676 cars per day. The volume of free air apparently compressed for this work was 279,200 cu.ft., of which 83.4% is accounted for, leaving 16.6% for leakage and slip in the compressor, leakage in air lines, and changes in temperature.

* *Transactions American Institute of Mining Engineers*, Vol. XXX, p. 566.

TABLE LIII

HAULAGE DATA, GLEN LYON COLLIERY

	Shaft Loco. No. 2 Plane.	Shaft Loco. No. 3 Plane	Slope Loco.
Number of trips, empty	3	10	16
Number of trips, loaded	3	10	15
Average number cars per trip, empty	15 33	12 7	11 4
Average number of cars per trip, loaded	13	13	11 3
Average cu ft. free air per trip, empty	1,724	5,686	1,230
Average cu ft. free air per trip, loaded	1,631	1,898	599
Average cu.ft. free air per round trip	3,355	7,584	1,829
Average cu.ft free air per ton-mile, on gross tonnage.	113		71
Average cu. ft free air per ton-mile, on net tonnage .	203		128

Cost of the plant, omitting boilers, was:

Compressor and extras		$2,955 75
Two locomotives and extras		5,869 76
Pipe-line. 5-in. line, 6,000 ft	$2,914 32	
3-in. line, 4,000 ft	1,240 46	
		4,154 78
Steam connections to compressor		278 27
Material and labor on compressor house and foundations, and installing pipe-line, etc		1,525 23
Charging stations.		372 21
Total cost		$15,156 00

Average cost of operation for 2 years, on basis of 170 days' work per year, was $12.60 per 10-hour shift, including $2.32 for steam for compressor, furnished by main boiler plant of mine. Adding proportion of fixed charges, with interest, depreciation and repairs, the daily cost (300 days' work per year) would be $18.52 per day. For the 2 years, the average cost per ton-mile was as shown in Table LIV.

In these two years the saving over the expense of the mule haulage, previously employed, was $14,218, or nearly the total cost of the haulage plant.

TABLE LIV

HAULAGE OPERATING COSTS, GLEN LYON COLLIERY

	1897 (179 DAYS)			1898 (160 DAYS).		
	Daily Ton Miles.	Daily Cost	Cost per Ton-Mile, Cents	Daily Ton-Miles.	Daily Cost	Cost per Ton-Mile, Cents.
Shaft locomotive, gross tonnage	1,485	$11 12	0 75	1,521	$12 00	0 79
Shaft locomotive, net tonnage	825	11 12	1 35	845	12 00	1 42
Slope locomotive, gross tonnage	648	11 12	1 72	720	12 00	1 67
Slope locomotive, net tonnage	360	11 12	3 09	400	12 00	3 00
Both locomotives, gross tonnage	2,133	22 23	1 05	2,241	24 01	1 07
Both locomotives, net tonnage	1,185	22 23	1 89	1,245	24 01	1 93

10. Following is the cost of a large colliery plant, as given by Beverly S. Randolph,* who installed and operated it:

Three-stage, compound compressor		$5,300
Pipe line. 5,600 ft., 5-in	$5,600	
3,100 ft, $3\frac{1}{2}$-in	1,700	
1,000 ft, $1\frac{1}{2}$-in	300	
		7,600
Two main locomotives, weight 30,000 lbs each		6,000
Five gathering-locomotives, weight 8,000 lbs each		10,000
Two boilers, each 80 H P		1,000
Installation, labor, and material		4,000
Total cost		$33,900

This plant includes an unusually large number of small gathering-locomotives. If the equipment had consisted of four 25,000-lb. engines, costing, say, $2,800 each, and which would do the same work, the total cost would be $29,100.

11. Inspiration Consolidated Copper Co., Arizona. In 1914 six 10-ton two-stage locomotives were installed for underground service; in August, 1917, there were 11 locomotives in use, including 2 spares. Over-all dimensions, 5 ft. 9 ins. high by 5 ft. 4 ins. wide by 16 ft. long. Rigid wheel-base, 4 ft.; track gage, 30 ins.; 4 drivers, 26 in. diameter; total weight, 20,000

* *Trans. Instn. of Min. Engrs.* (England), Vol. XXVII (1904), p. 433.

lbs., all on drivers; drawbar pull, 4,400 lbs.; cylinders, 7 ins. and 14 ins. by 14-in. stroke; locomotive storage tank, 40-in. diameter by 12 ft. 8 ins. long, 105 cu.ft. capacity.

Air at 1,000 lbs. is furnished by two 4-stage compressors, each having a capacity of 1,125 cu.ft. of free air per min. Main air-line in shaft is 6 in., with 3-in. and 2-in. branches to charging stations; there are no receivers. Average charging time, 1 min. Charging pressure, 800 lbs., which is dropped by a reducing valve to 250 lbs. in an auxiliary reservoir. In the high-pressure cylinder, the air is expanded to 50 lbs., causing a temperature drop to about 140° F. below normal. Before the air enters the low-pressure cylinder, its normal temperature is approximately restored by an interheater.

Trains average 15 cars (maximum, 25 cars), each weighing 2 tons and carrying 5 tons of ore; train crew, engineer and 2 helpers; 5 cars at a time are dumped in a rotary tipple. Average haul (August, 1917), 0.475 mile. Locomotive is charged once per round trip. Average number of trips, 15 per 7 hours of actual hauling time. Power per ton-mile, 0.796 kw. Cost of haulage is variable, due to sliding wage scale.

CHAPTER XXVII

MEASUREMENT OF AIR CONSUMPTION

A DISCUSSION of the transmission of compressed air through long pipes, with formulas and numerical data, is given in Chapter XVI. The behavior of air is different when flowing through short pipes, or orifices in thin plates, and under pressures relatively small as compared to those existing in ordinary compressed-air transmission (see page 222).

Flow of Air through Orifices or Short Tubes. The fundamental equation is $v = c\sqrt{2gh}$, in which: v is the velocity of flow, feet per second; g, the force of gravity, or 32.2; h, the height in feet of a column of air, or head, required to produce the pressure under which the air is flowing, that is, $h = p^1 - p^2$ = difference between the pressures on the two sides of the plate containing the orifice; and c, a coefficient of flow, determined experimentally. The volume discharged, in cubic feet per second, is equal to v multiplied by the area of the orifice in square feet.

Values of constant "c"

(Weisbach, Mechanics of Engineering, p. 945–947).

FLOW THROUGH AN ORIFICE IN A THIN PLATE

Diameter of orifice = 1 cm. = 0 394 inch:

Ratio of pressures on the two sides of the orifice $= \frac{p^1}{p^2}$	1 05	1 09	1 43	1 65	1 89	2.15
Values of c	0 555	0 589	0 692	0 724	0.754	0 788

Diameter of orifice = 2 14 cm. = 0 843 inch:

Ratio of pressures	1.05	1 09	1 36	1.67	2 01
Values of c	0 558	0.573	0.634	0.678	0.723

FLOW THROUGH A SHORT TUBE

Diameter of tube = 1 cm. = 0.394 inch; length 3 cm. = 1.181 inch:

Ratio of pressures $= \frac{p^1}{p^2}$	1.05	1.10	1.30
Values of c	0.730	0.771	0.830

Diameter of tube = 1.414 cm. = 0.557 in., length, 4.242 cm. = 1.670 inch:

Ratio of pressures	1.41	1.69
Values of c	0.813	0.822

Diameter of tube = 1 cm. = 0.394 in., length, 1.6 cm. = 0.630 in. Edge of orifice rounded.

Ratio of pressures	1.24	1.38	1.59	1.85	2.14
Values of c	0.979	0.986	0.965	0.971	0.978

In this case c always approximates unity. A short conical pipe, rounded off at the inlet orifice, gave nearly the same values for c.

An empirical formula for the velocity of flow, suitable for dealing with the small differences of pressure used in ventilation, is (Clark's Rules, Tables and Data, p. 891):

$$V = C\sqrt{\frac{2gh}{12} \times 773.2 \times \left(1 + \frac{t-32}{493}\right) \times \frac{29.92}{p}}$$

$$= 352C\sqrt{[1 + 0.00203(t-32)]\frac{h}{p}}$$

where: V = velocity of flow, feet per second; C = coefficient of efflux; $g = 32.2$; h = inches of water column measuring the difference of pressure; t = the temperature, Fahr.; p = barometric pressure, in inches of mercury. The constant 773.2 is the volume of air at 32°, under a pressure of 29.92 inches of mercury, when the volume of an equal weight of water is taken as 1.

For 62° F., $V = 363C\sqrt{\frac{h}{p}}$, and if $p = 29.92$, $V = 66.35C\sqrt{h}$.

In this formula, the values of C (Weisbach), for pressures of 0.23 to 1.1 atmosphere, are:

For circular orifices in thin plates................ 0.56 to 0.79
For short cylindrical tubes .... 0.81 to 0.84

For the same, rounded at the inner end. . ... 0.92 to 0.93
For conoidal mouthpieces, in form of the "vena contracta". . . . . 0.97 to 0.99
For conical converging mouthpieces . . . 0.90 to 0.99

TABLE LV

ACTUAL DISCHARGE, POUNDS OF DRY AIR PER SECOND, AT 60° F. AND 14.7 LBS. BAROMETRIC PRESSURE, FOR CIRCULAR ORIFICES IN A PLATE 0.057 IN. THICK (1 LB. OF AIR UNDER THE ABOVE CONDITIONS = 13.09 CU. FT.).* Kent.

Pressure, Inches of Water Column †	Diameter of Orifice, Inches								
	0 3125	0 500	1 000	1 500	2 000	2 500	3 000	3 500	4 000
½	00114	0 0203	0117	0203	0408	073	105	143	187
1	00162	00416	0166	0373	0663	103	149	202	264
1½	00199	00510	0203	0457	0811	127	182	248	323
2	00231	00590	0235	0538	0937	146	210	285	373
2½	00259	00662	0263	0591	1050	163	235	319	416
3	00285	00726	0289	0648	1150	179	257	349	455
3½	00308	00786	0312	0700	1240	193	277	377	491
4	00330	00842	0334	0749	1330	206	296	402	525
4½	00351	00895	0355	0794	1410	219	314	426	556
5	00371	00945	0375	0838	1480	231	331	449	586
5½	00390	00993	0393	0879	1550	242	347	471	613
6	00408	01040	0411	0918	1620	252	362	492	640

* The general expression for the weight of dry air is $W = \frac{1.325B}{T}$, where W = weight of 1 cu ft of air, B = absolute pressure, inches of mercury; T = absolute temperature (F.).
† 1 inch of water column = 0 0361 lb per sq in = 5 20 lbs per sq ft

Measurement of Low-pressure Air. Fig. 247 shows an apparatus for measuring small quantities of air at low pressure.* It can be used for any ordinary working pressure, as employed for mine service, but the actual measurement is applied to an equal flow of air under a low pressure. That is, for convenience in measuring while carrying on the test, an equal volume of flow of low-pressure air is substituted for the regular high-pressure

* G. S. Weymouth and C. C. Freeman. *Jour. Chamber of Mines*, Kalgoorlie, Western Australia, 1912.

supply. The apparatus here described is for dealing with small volumes, say 20 cu. ft., or less, of free air per minute; but, by using larger tanks, any quantity can be measured, provided the supply pressure is constant during the test and the flow regular for the few seconds necessary to obtain the first gage reading. The volume of air thus isolated is then measured at leisure. In using this method, the regular supply must be diverted for about 0.5 minute.

Connected with the air supply is a vertical 1-in. pipe, with globe-valves *A* and *D*, stop-cock *B*, and a mercury gage *I*. This pipe leads by a hose to a closed water tank *K*, of about 30 cu. ft. capacity, which is provided with a gage glass and a short open hose *H* from the bottom. A branch pipe, with stop-cock *C* and globe-valve *E*, leads to a 5-gal. drum (of light iron or tin plate) connected with which is a water-gage *J*. The drum has an opening at *F*, to which disks with orifices of different sizes can be applied, the area of orifice being such as to keep the pressure in the drum low enough to eliminate factors caused by variations of volume due to pressure. These orifices may be of, say, $\frac{1}{8}$, $\frac{1}{4}$, $\frac{1}{2}$ and $\frac{3}{4}$ sq. in area.

To make a test, valve *A* is opened to admit the necessary quantity of air; cock *C* is then opened and valve *E* regulated to give any required pressure on the mercury gage *I* (for example, 5 lbs. per square inch). An orifice is attached at *F* to the 5-gal. drum, to give a reading on the water-gage *J* of 2 to 9 ins. (=0.0722 to 0.325 lb. per square inch). The hose at *G* is disconnected, cock *B* opened, cock *C* closed, and valve *D* regulated to give the same reading on gage *I* as before; thus, the resistance of valve *D* is made equal to that of valve *E*, leading to the orifice at *F*. Tank *K* is then nearly filled with water, and its level noted on the gage glass; cock *C* is opened, *B* is closed, and the hose is connected at *G* to tank *K*. Next, cock *B* is opened and *C* closed simultaneously, and the free end of hose *H* lowered; the reading on gage *I* is noted and, when the tank is nearly empty, *C* is opened and *B* closed simultaneously. The time the air is passing to tank *K* is taken by stop-watch; the air in *K* is reduced to atmospheric pressure (by bringing the free end of hose *H* to the

water level in the tank), and the volume of air noted. If necessary, due to any back pressure from tank K, gage I is brought to the same level as when air was passing to K, by manipulating valve E, and a reading on J is taken.

Table LVI shows the results of measurements, using orifices of 0.214, 0.441 and 0.797 sq. in., a pressure on the mercury gage

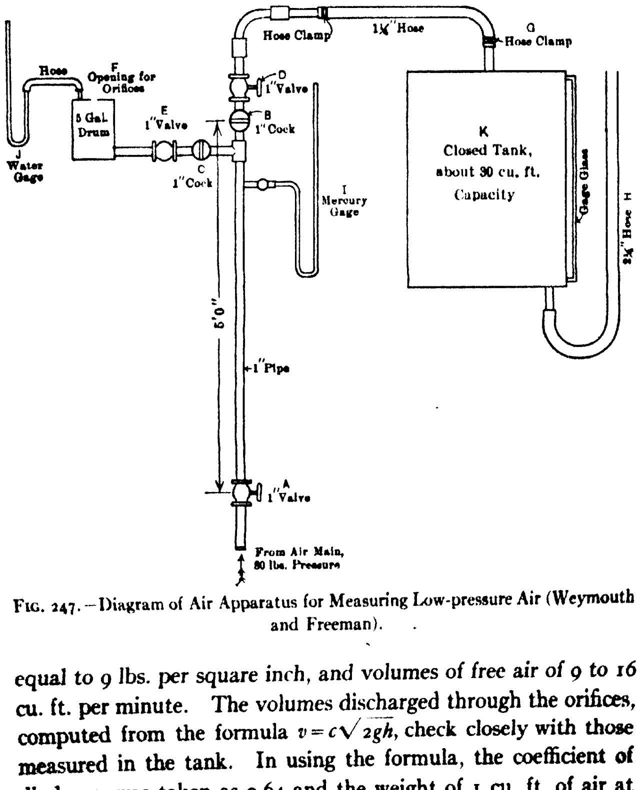


FIG. 247.—Diagram of Air Apparatus for Measuring Low-pressure Air (Weymouth and Freeman).

equal to 9 lbs. per square inch, and volumes of free air of 9 to 16 cu. ft. per minute. The volumes discharged through the orifices, computed from the formula $v = c\sqrt{2gh}$, check closely with those measured in the tank. In using the formula, the coefficient of discharge was taken as 0.64 and the weight of 1 cu. ft. of air at 62° F., as 0.0671 lb.

Measurement of High-pressure Air. (1) For comparatively small volumes, the air delivered by a compressor may be passed

into a tank or receiver of known capacity, provided with a pressure gage; (2) the air may be measured by causing it to displace water in a tank; (3) the air may be metered.

TABLE LVI

RESULTS OF MEASUREMENTS OF LOW-PRESSURE AIR BY APPARATUS OF G. S. WEYMOUTH AND C. C. FREEMAN. (*Mining Press*, April 20, 1912).

Area of Orifice F, Sq In	Mercury Gage I, Ins	Water Gage J, Ins	Calculated Volume, Coeff of Disch = 0.64		Volume Passed into Tank, Cu Ft	Time, Secs	Volumes of Free Air in Tank	
			Cu Ft per Sec	Cu Ft per Min			Cu Ft per Sec	Cu Ft per Min
0.214	12	$8\frac{9}{16}$	0.184	11.0	17.40	96.5	0.181	10.9
0.214	19	$9\frac{9}{16}$	0.194	11.6	23.01	108.0	0.212	12.7
0.214	16	$7\frac{1}{16}$	0.167	10.0	16.11	94.5	0.171	10.3
0.214	16	$5\frac{7}{8}$	0.152	9.1	13.82	88.5	0.156	9.4
0.214	16	$5\frac{3}{4}$	0.151	9.1	13.82	88.0	0.157	9.4
0.441	9	$2\frac{1}{4}$	0.194	11.6	19.33	96.0	0.201	12.1
0.441	12	$3\frac{1}{8}$	0.237	14.2	20.48	86.0	0.238	14.3
0.441	13	$1\frac{15}{16}$	0.186	11.2	22.55	116.0	0.194	11.6
0.441	16	$4\frac{3}{8}$	0.271	16.3	22.86	87.0	0.263	15.8
0.441	16	$4\frac{1}{4}$	0.267	16.0	19.40	74.0	0.262	15.7
0.797	12	$\frac{15}{16}$	0.228	13.7	20.48	86.0	0.238	14.3
Aver'g's			0.203	12.2			0.207	12.4

Measurement of Discharge through Orifices. This is as accurate as the use of tanks, and is applicable to both small and large volumes. The compressed air is discharged through orifices of known area, and the equivalent volume of free air computed.

One apparatus for this method consists of a short piece of wrought-iron pipe, in which are set a number of short branches, each provided with an orifice, the orifices being of different diameters.* Each branch has a valve, the whole resembling a manifold as used for connecting a group of rock drills to an air main. Fig. 248 shows a meter with 8 orifices, ranging from $\frac{3}{32}$ to $\frac{5}{8}$ in.

* F. D. Holdsworth, Chief Air-Compressor Engineer, Claremont plant, Sullivan Machinery Co. *Eng. and Min. Jour.*, May 25, 1912, p. 1028.

diameter. The orifices are accurately reamed holes in disks of steel plate, held by flanges in the outer ends of the branch pipes. For the larger orifices the disks are $\frac{1}{2}$ in. thick; for the smaller, $\frac{3}{8}$ in. The back or pressure side of each hole is rounded to a radius $\frac{1}{16}$ in. less than the thickness of the plate. The actual diameter of the finished hole is measured by a micrometer and the area calculated.

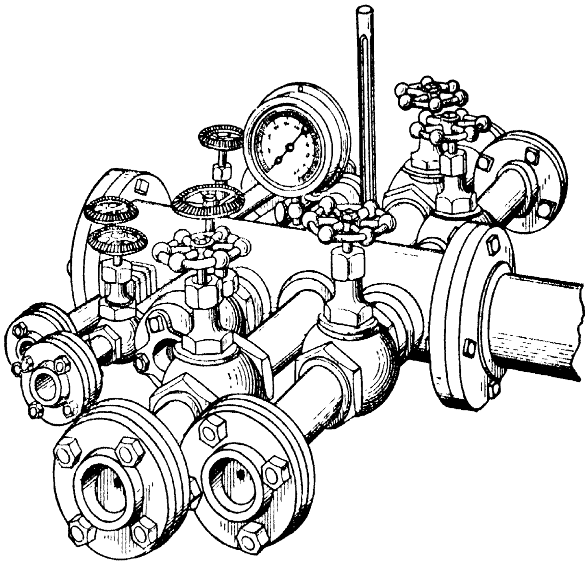


FIG. 248.—Manifold and Nozzles for Measuring the Discharge of Compressed Air (*Eng. and Min Jour.*, May 25, 1912)

The rate of flow through an orifice of this form is obtained from Fliegner's formula,

$$G=0.53\frac{AP}{\sqrt{T}}$$

where G = flow, pounds per second; A = area of orifice, square inches; P and T = absolute pressure and temperature (F.) of

the air behind the orifice. For the formula for the weight of a cubic foot of air, see the footnote under Table LV.

Several small orifices are used instead of a single large one, because, though the formula is accurate for orifices not greater than $\frac{5}{8}$ in., it is inaccurate for larger ones. Also, by having a series of orifices of different diameters, any desired combination may be made according to the volume of air to be measured. Tapped into the main pipe of the manifold is an accurate pressure gage and a well for a thermometer, which, for a 2-stage compressor, should read to at least 300° F.

To prepare for a test, disconnect the air main near the receiver and attach the manifold. Stop all other outlets from the receiver, so that all the air entering the receiver will pass through the orifices. Place another thermometer in the compressor intake, close to the cylinder. The compressor speed is recorded by a revolution counter. Determine by trial the proper combination of orifices to maintain the desired pressure. Run the compressor about 2 hours, discharging through the orifices, so that both pressure and temperature will become maximum and constant.

One observer begins recording the compressor revolutions, and another, at 1-minute intervals, takes the temperature and pressure readings at the orifices; at the same time, the intake temperature is also recorded. In 10 or 15 minutes enough temperature and pressure readings will be obtained, and a final reading is made of the revolution counter, to find the total number of revolutions for the period of the test. The total piston displacement is then computed. Finally, the quantity of air discharged through the orifices, determined by the formula and divided by the total displacement, gives the volumetric efficiency.

For accurate results, obtain the local barometric reading, from the nearest Weather Bureau station, or otherwise. This, with the temperature at the compressor intake, is used in the formula, $W = \frac{1.325B}{T}$ (see note under Table LV), for reducing to cubic feet of free air per minute the pounds of air per second,

computed by Fliegner's formula. In testing a stage compressor, only the displacement of the low-pressure cylinder is used, since this measures the volumetric capacity of the compressor.

A method of measuring air by pitot tubes was adopted at a mine in northern Michigan, for finding the quantity of air used in a large blacksmith shop.* The compressor was working at a pressure of 75 to 80 lbs.

Two pitot tubes, made of copper tubing $\frac{1}{16}$ in. outside diameter, were soldered into a $\frac{3}{8}$-in. to $\frac{1}{2}$-in. bushing. This bushing in turn was screwed into a hole tapped in the side of a piece of 1-in. iron pipe, 20 ins. long, which was inserted in the air main. The pitot tubes projected $\frac{1}{4}$ in. into the pipe; one of them being directed toward the air current, the other set at right angles to it. They were connected to a U-tube, made of two gage glasses joined at their lower ends by rubber hose. Either a water or a mercury gage may be used to determine the velocity head of the air current. The 20-in. piece of 1-in. pipe was provided with a thermometer and pressure gage, set behind the pitot tubes and at a little distance from them, to avoid producing eddies.

The volume of air flowing through the 1-in. pipe was computed by the formula

$$Q = 135\sqrt{\frac{H(P+14.7)}{t+460}},$$

where Q = cubic feet of free air per minute, at 60° F. and 28.5 ins. barometer; H = height of water column, inches, representing the velocity head (if mercury be used, multiply the reading by 13.6, which is the specific gravity of mercury); P = reading of pressure gage; t = thermometric temperature of the compressed air (degrees F.).

To furnish some storage capacity, and so prevent sharp peaks in the flow of air caused by the intermittent use of the drill sharpener and the shop hammer, the measuring device was set in the air line at a little distance from the shop. Measurements were made on a Saturday afternoon, when air was being used only for the shop and rock-houses. The result showed: total

* B. B. Hood, *Eng. and Min. Jour.*, June 27, 1914, p. 1283.

free air compressed, 670,000 cu. ft.; blow-off, 338,000 cu. ft.; line loss, 118,000 cu. ft.; blacksmith shop, 27,000 cu. ft.; rock-houses, 187,000 cu. ft. The high line loss was largely due to expansion and contraction in the spiral-riveted pipe and its joints. Most of the leakage was in the 500 ft. of air main nearest the compressor, where the variations of temperature were naturally greatest. Underground, the piping was found to be practically tight.

"Ready-made" Meters of several forms are on the market; for example, the "Tool-om-eter," made by the New Jersey Meter Co., Plainfield, N. J., and the instrument of the Bailey Meter Co., Boston.

The Bailey meter works on the principle of measuring accurately the pressure difference across an orifice placed between a pair of flanges in the pipe line. A special orifice plate is used, made of $\frac{1}{32}$-in. Monell metal. The area of orifice is proportioned to the size of pipe, so that, at average rates of flow, a pressure-drop of about $\frac{1}{2}$ lb. takes place. The record is automatically made on a direct-reading chart, with uniform graduations. (For a detailed description, see *Met. & Chem. Eng'g.*, April 15, 1916, p. 456.)

The "Tool-om-eter" (Fig. 249) is a small, simple direct-volume gage, employing the principle of multiple nozzles. It has but one moving part, which consists of the weighted piston *a*, in the metering cylinder *b*, and a piston rod *c*, which carries a small piston *d* in the oil dashpot *e*. An extension of the piston rod above *a* moves freely, without contact, inside of the sight glass *f*, and the height of its upper end is read on the accompanying scale, which records cubic feet of free air per minute.

At *g* air enters the space around the dashpot cylinder *e* and the oil reservoir, and then passes through ported openings into the metering cylinder *b*. Through the walls of this cylinder are drilled a large number of small holes, accurately spaced and reamed. To pass to the outlet chamber *h*, the compressed air lifts piston *a*, thus exposing some of the holes to the flow. The air leaves the meter at *i*.

The small "head" or difference of pressure (a few ounces per

square inch), produced between the interior of cylinder *b* and the chamber *h*, is fixed by the relation of the weight of the moving element to the area of the piston *a* on which the difference of pressure acts. That is, the moving element rises until its weight is exactly supported by the difference of pressure; so that the pistons and rod float in static balance, in a position corresponding to the volume of air flowing and the number of holes exposed in cylinder *b*. The scale is calibrated by comparison with a standard instrument.

This meter is made in two sizes: the smaller for flows of 10 to 100 cu. ft., and the larger for 50 to 300 cu. ft. of free air per minute.

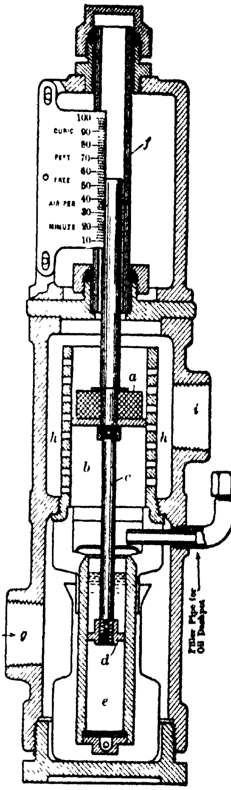


FIG. 249. –The "Toolom-eter," for Measuring Compressed Air.

It may be suggested that the water meter made by the Worthington Pump & Machinery Co. might readily be modified for measuring compressed air. The measuring chamber of this meter contains a "wabbling disk," resting at its apex on a ball, and set between the inlet and discharge openings. A slit cut in one side of the disk fits a radial septum or division in the disk chamber; water is admitted on one side of the septum and discharged at the other side. The water pressure causes the disk to make a uniform wabbling motion, which passes water continuously through the meter; though there is at no time any free opening, through which water could flow without moving the disk. A short rod, attached at right angles to the upper side of the disk, communicates motion to a gear-train, which operates a counter registering cubic feet of water on

a dial. This meter, like some others, is of the "inferential" type; that is, it must be rated to determine its discharge.

Measuring Tanks. Two large vertical tanks are generally used, connected with the compressor and with a pump for supplying water to the tanks, long vertical gage glasses being provided for reading the water level. The air in the two tanks is always separated by a piston-like mass of water, which flows back and forth from one tank to the other. The change in level, multiplied by the area of the tank, is equal to the volume of air that has flowed out. In order that the observed differences of water level shall accurately represent corresponding volumes of air, it is evident that the internal cross-sections of the tanks must be constant throughout their heights.

The machine to be tested is connected to one tank, both tanks are filled with air at the desired pressure, and the pump keeps the air pressure constant by forcing in water. On starting the drill, or other machine, the height of water in the gage glass is noted. After a run of sufficient length to secure accuracy, the drill is stopped and the height of water in the gage glass on the tank which has been used is again noted. The increase in volume of water in the tank is the volume of compressed air used during the run; from this the volume of free air is computed. Having two tanks, one is used while the other is being filled with air, thus allowing tests to be run for any desired time.

Fig. 250 shows an apparatus, which though more elaborate is practical and convenient, and the tanks need not be of uniform cross-section throughout.* Instead of using long gage glasses, reaching from top to bottom, one tank has two short glasses, *j* and *h*, one at top, the other at the bottom; the companion tank has a single gage, at the top.

As the tank air must be kept at constant pressure, a reducing valve is provided (the valve known as the "*S*-4 locomotive governor," made by the Westinghouse Air Brake Co., is recommended). It is arranged as at *a*, Fig. 250. The control pipe *b* should not be connected directly to either tank, as the valve continuously exhausts air to the atmosphere. A

* Walter S. Weeks, *Mining Press*, December 15, 1917, p. 855.

lubricator *r* is placed on pipe *b*, for feeding a thin oil to the valve.

To transfer the air from one tank to the other, a 4-way valve

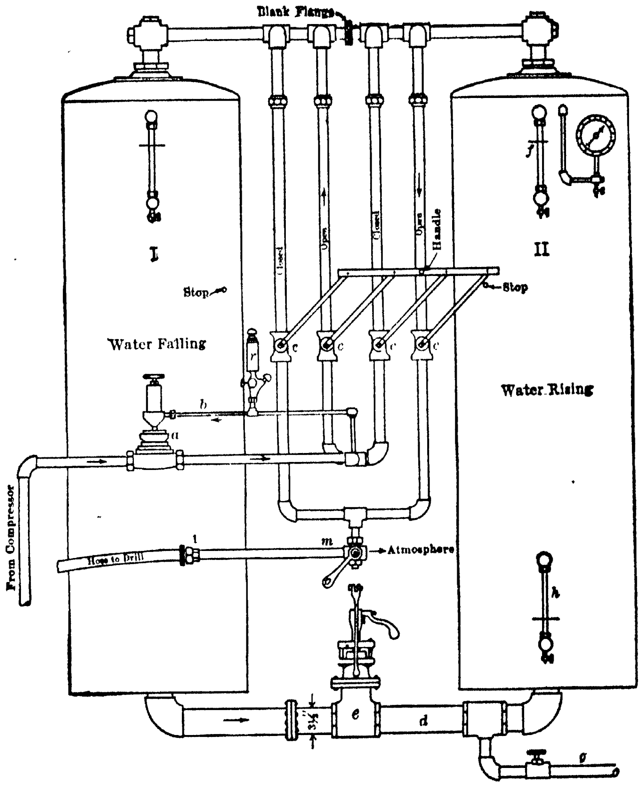


FIG. 250.—Measuring Tanks for Rock-drill Testing (W. S. Weeks, *Mining Press*, December 15. 1917).

is generally used (see Fig. 251). It has the disadvantage of being liable to leakage. Fig. 250 shows an arrangement of 4 pipes, each provided with a single, 1-in. cut-out valve *c*. The valve levers are connected, so that all are thrown together, thus pro-

ducing the effect of a 4-way valve. In pipe *d*, connecting the tanks at the bottom, is a quick-closing gate-valve *e*.

The volume of one tank only needs to be measured, that having the two gage glasses. First place a mark on glass *f*, and fill the tank with water to that point through pipe *g*. Close valve *e*, and draw off water through pipe *g* until the water level is sighted on glass *h*. Then carefully draw off more water, until a convenient even number of cubic feet is reached, and mark the corresponding point on glass *h*. Thus, when in operation the water rises from the lower to the higher mark, the measured volume of air has been displaced.

To test a drill, connect the hose at *l*, as shown; then fill tank I with water; close the gate-valve *e*; bring the water level in tank II to the lower mark; open the gate-valve and throw valves *c* to the position shown, thus forcing water from tank I to tank II until it reaches the upper mark; then close the gate-valve. The reducing valve *a* is adjusted to hold the pressure constant, notwithstanding the small changes in hydraulic head on the air. When necessary, the air in the tanks may be exhausted to the atmosphere through the 3-way valve *m*. The drill test may be continued as long as desired by running the water a number of times from one tank to the other. For ordinary testing, the volume between the gage-glass marks on tank II should be 16 or 18 cu. ft.

A less expensive apparatus consists of two small tanks, say 2 ft. diameter by 4 or 5 ft. high, connected at the bottom by a 3-in. pipe and provided with gage-glasses (Fig. 251).* The tanks are about half filled with water. Pipes from the top of each tank are connected to a common 4-way valve. With the valve in the position shown in the cut, water in the right-hand tank is forced by the air pressure into the left-hand tank, and the air in the latter is driven out through the valve to the drill. When the water level reaches a definite point near the upper end of the gage-glass on the left-hand tank, the valve handle is quickly thrown to the position shown by dotted lines. The

* George H. Gilman, Sullivan Machinery Co., *Compressed Air Magazine*, Aug. 1912, p. 6510.

process is thus reversed, the air in the right-hand tank then passing to the drill.

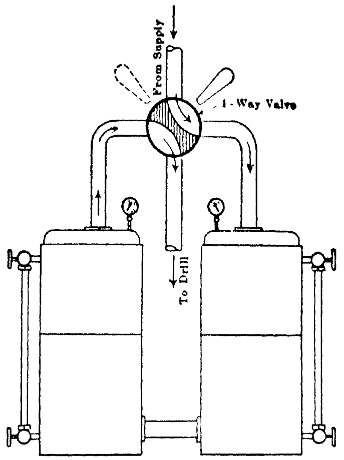


FIG. 251.—Tanks for Measuring Air Consumption of Rock Drills.

The total volume of compressed air used is found by counting the number of times each tank is filled and emptied during a given time, and the corresponding volume of free air is computed as already explained.

Measuring Air Discharge with One Tank.* An accurate method is as follows. To one end of the air receiver is connected

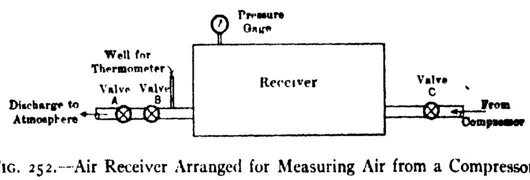


FIG. 252.—Air Receiver Arranged for Measuring Air from a Compressor.

the air pipe from the compressor. At the other end is a horizontal discharge pipe, provided with two globe valves set close together and a well for a thermometer (Fig. 252). This pipe

* E. E. Fessenden, *Compressed Air Magazine*, February, 1913, p. 6714.

may be a part of the air main, disconnected from the service line so as to discharge into the atmosphere. On the receiver is an accurately calibrated pressure gage.

To begin a test, first open valves *A* and *B* wide; then, with the compressor at normal speed, gradually close valve *B* until the tank pressure is constant. After a short run, to insure that pressure and temperature conditions have become constant, close valve *A* tight and allow the tank pressure to rise to 10 or 15 lbs. above normal working pressure. Then close valve *C*, stop the

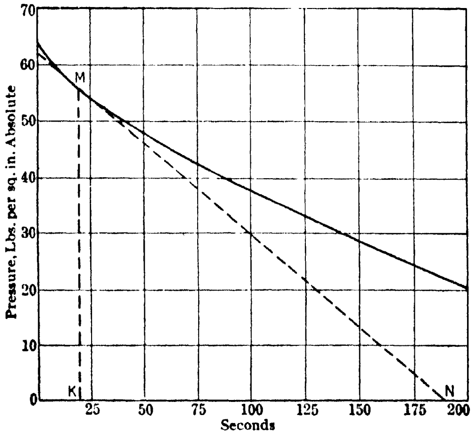


FIG. 253.—Time-Pressure Curve.

compressor, open valve *A* quickly, and take exact simultaneous readings of time and pressure, as the tank pressure falls due to escape of air through valve *B*. An assistant holding a watch should call "read" at uniform intervals of 10 or 15 seconds, continuing until the pressure has dropped considerably below the working pressure. Finally, plot the readings on cross-section paper, to obtain a time-pressure curve (Fig. 253).

From Chapter III, p. 50, if the weight of the volume of compressed air in the tank $=W$, then $PV=WRT$, whence $W=\frac{PV}{RT}$. *W* is taken as the entire volume of the tank and its connections

between valves B and C; it is determined by weighing the volume of water required to fill the tank and connections, and dividing by 62.355 (weight of 1 cu. ft. of water at 62° F.).

Differentiating the above equation with respect to time: $\frac{dW}{dt}=\frac{V}{RT}\times\frac{dP}{dt}$, in which $\frac{dW}{dt}$ is the rate of decrease of weight of air in the tank, or the rate of discharge in pounds per second, while $\frac{dP}{dt}$ is the rate of decrease of pressure.

In Fig. 253, M is a point on the time-pressure curve corresponding to the normal working pressure. MN is carefully drawn tangent to the curve at M, and MK is parallel to the pressure axis. Then $\frac{dP}{dt}=\frac{MK}{KN}$. Substituting in the differential equation: $\frac{dW}{dt}=\frac{V}{RT}\times\frac{MK}{KN}$. Reading from the diagram the values of MK and KN, the rate of discharge, in pounds per second, is found directly. As the weight of air discharged equals the weight drawn into the compressor, the free air capacity is found from $V_0=\frac{WRT_0}{P_0}$, where V_0 is the volume, in cubic feet of air per second, at the intake pressure P_0 (pounds per square foot absolute) and at the absolute temperature T_0, and W is the weight of air discharged per second as already found.

Another mode of using a single tank, especially adapted to testing rock-drills, is shown in Fig. 254. A vertical tank or receiver is connected to the main air line from the compressor by a 1-in. pipe, another 1-in. pipe leading to the rock-drill. On the receiver is a line of overlapping water-gages, or a single long gage glass, extending from bottom to top. Connected to the bottom of the receiver is a 3-in. pipe from a pump (or a water tank at an elevation sufficient to give a static pressure equal to the air pressure).

To begin a test, run in enough water to show at the bottom of the gage glass, and make a chalk mark opposite this point. Next, turn on the compressed air until the receiver is filled at the desired pressure; then turn off the air and start the drill. During

the run, maintain a constant-pressure in the receiver, as indi cated on the pressure gage, by regulating the valve in the water pipe. When the drill is stopped, mark instantly the height of water level on the gage glass by another chalk mark. Note the distance between the marks, and the elapsed drilling time. The cross-section of the receiver being known, compute the volume of compressed air used, and the corresponding volume of

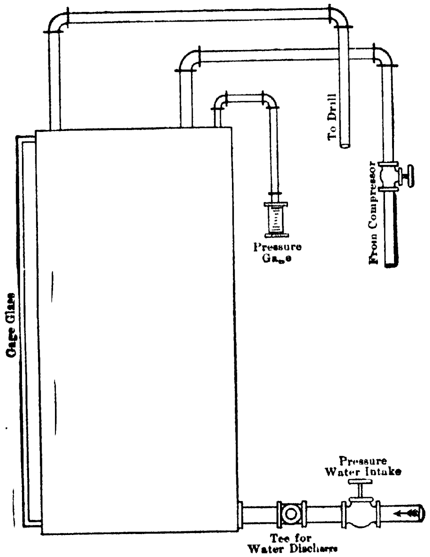


FIG. 254.—Diagram of Tank for Measuring Air Consumption of Rock-Drills.

free air. From the depth of hole drilled the air consumption per linear inch or foot of hole may also be found.

If it be assumed that the receiver pressure is constant, the volume V of free air used is

$$V=RA\left(\frac{P+H}{H}\right)$$

where R=rise of water level, feet; A=cross-section of receiver, square feet; P=initial gage pressure, pounds; and H=atmospheric pressure.

In case the receiver pressure decreases from P to P' during the test, let V = initial volume of free air in the receiver at pressure P, and V' = final volume of free air at pressure P'. Then

$$V = AB\left(\frac{P+H}{H}\right) \text{ and } V' = A(B-R)\left(\frac{P'+H}{H}\right),$$

where A, R and H are as above, and B = total height from initial water level to top of receiver, feet. Then the volume of free air used $= V - V'$.

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